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# Naval ship control systems.

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# THESIS

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September 1970

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# ABSTRACT

An analysis of the problem of Naval Ship Automation is made. The purpose is to search the nonclassified existing documents and to present a general view of the state of the art in Naval Ship Control Systems. This study covers material that is concerned with conventional Naval Surface Ships such as Destroyers and Frigates. Fire Control Systems are not considered because such studies already exist at the present time.

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# I. INTRODUCTION

Broadly defined, "automation" is the total control of a process by automatic means using automatic control systems without the necessity for human intervention. Automation of ship machinery plants is potentially very attractive to the Navy, since it should result in a substantial decrease in shipboard operating personnel, improved operating efficiency, greater reliability and safety, and improved controllability.

Totally automated systems have been built for controlling processes which equal or exceed the complexity of a shipboard machinery system. While technically possible, total automation of naval ship machinery is not a realistic goal, now or in the near future. There are two reasons for this. First, it would be prohibitively expensive. When a man is acting as a simple follow-up servo, his function can be mechanized relatively simply and cheaply. On the other hand, when the man is a decision element operating on subjective data, mechanization is extremely complex and costly. Secondly, total automation is incompatible with the mission of a combatant ship. Such a ship is intentionally operated in an environment made hostile by the actions of a malicious enemy. It must expect to sustain damage and damage induced casualties, and be designed to survive and retain a reasonable probability of mission accomplishment. This does not mean that any "automated" feature

which may fail under these circumstances must be rejected. It does mean that an orderly, efficient system of "fallback" control modes must be provided. It is precisely this procedure with a near infinite range of situations and responses which is almost impossible to automate. When any portion of the machinery is operable, there must be means for controlling it. The last step in the "fallback" control arrangement is complete manual operation of at least enough machinery to keep the ship under way and effective. This sets a lower limit on the operating personnel, and consequently an upper limit on the justifiable extent of automation.

Complete automation requires that all control loops be closed through one or more central decision elements. Hereafter, this device(s) will be loosely termed a "computer." The computer must sense all required plant conditions and variables, and produce the proper machinery commands. This is not particularly difficult, if it can be assumed that the machinery and control system will function perfectly at all times, and that all possible operating conditons can be foreseen; i.e., there is no requirement for judgement, adaptability, or subjective decisions. Such an assumption in the case of a combatant ship machinery plant is both highly unrealistic and dangerous.

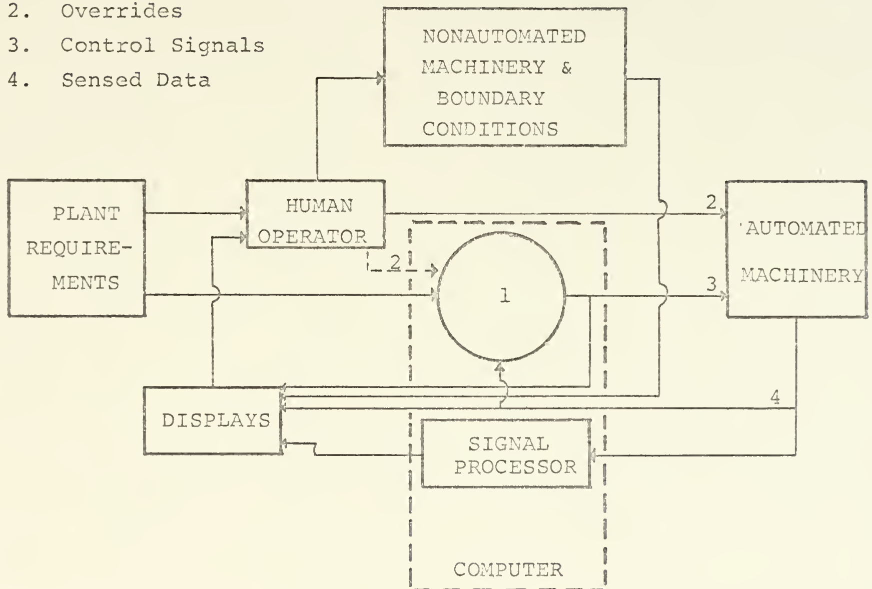
The approach to "automation" currently being applied extensively in the merchant marine, involves closing most control loops through a human operator. However, the

instrumentation required for control, and actuators for the controlled machinery are grouped in a central location, thereby requiring a minimum number of operators. This scheme is more appropriately termed "centralized control" rather than "automation."

A practical approach to naval automation is to utilize automatic control systems and a computer where their capabilities exceed those of a man and to employ man for judgement, evaluation and override on the whole system. Such an approach can produce savings in manpower and improve plant performance without the penalty of total dependence on perfect functioning of the system at all times.

In this system, the above approach has been implemented by a combination of centralized control with human interpretation of conditions and the issuing of commands together with fully automatic control in closed loop systems where conditions are sensed, interpreted, and commands are issued to the automated machinery without human intervention. Such a system is shown in block diagram form in Figure 1. From this diagram, it can be seen that the "computer" is interpreted in a broad sense to include both a signal processor (digital computer) and devices or circuits which provide automatic evaluation and issue commands based on data sensed from the automated machinery. The fully automated part of the system operates in a closed loop under the direct control of the "computer" which provides a status display to the man. The actions of this loop are monitored and when necessary, overridden by the man but not normally

1. Automatic Evaluation & Command
2. Overrides
3. Control Signals
4. Sensed Data



controlled by him. Such subsystems as an automatic turbine starting/shutdown programmer, electric plant start-stop-reversing controller, turbine control systems and automatic turbine shutdown equipment fall in this category.

The second control loop is not fully closed around the computer. Sensed data is processed for display to the man who then uses human judgement to take the necessary actions by initiating commands to the controlled machinery. The man then becomes part of the total automated plant. In this loop are made those decisions and commands relating to boundary conditions (relative wear or history of the equipment, which standby to use, etc.). In addition, decisions of a marginal or subjective nature are made and appropriate control action taken in this loop. This area includes initiation of casualty control procedures, override of the automatic loop functioning when circumstances require it, and initiation of actions which are carried out automatically but require manual starting, such as base turbine starting, control transfer, etc.

Also included in this loop are a number of decisions and control sequences which, while of a substantially routine nature, have not been mechanized due to their infrequency or their high cost of mechanization relative to its benefit. For example, most auxiliary machinery systems such as electric power, water and pneumatic systems are in this group.

In deciding if and to what extent a subsystem should be automated, the following general criteria have been applied:

1. Complexity of the automated system. (Reliability is assumed to be inversely related to complexity.)

2. Cost of the automation relative to its benefit.

3. Advantages of automation.

4. Manning reduction. Improved or more consistent operation. Safety.

5. Need for human judgement.

6. Damage and casualty control.

With these facts in mind, it is the purpose of this thesis to search the non-classified documents existing and to present a general view of the state of the art in Naval Ship Control Systems.

This study pretends to cover the material that is concerned with conventional surface naval ships such as destroyers, frigates and auxiliaries.

Fire Control Systems will not be considered in this thesis because such studies already exist at the present time.

## II. STEERING ENGINE

### A. PROBLEM

In the recent past we have seen the development of effective automatic steering control equipment and the continuing research into hull response and turning behavior. These are evolutionary steps forward in the solution of the problem of providing the most effecient directional control for ships.

Yet in this same period there has been relatively little change in the steering engine itself. This stagnation has been due mainly to the emergence many years ago of the electro-hydraulic system as markedly superior to all of the other then existing types. The resulting well justified reputation of excellent performance and reliability has generated not only a reluctance to change but also a reluctance to investigate and develop new systems. Certainly the first of these is understandable, but the second cannot be logically defended. It was challenged by Butterfield more than ten years ago; and since then the rotary vane, one of the systems that he recommended, has gained some acceptance, particularly abroad. More recently we have seen the development of a new high pressure rapson slide system in England and a rack and pinion electro-hydraulic system in the United States.

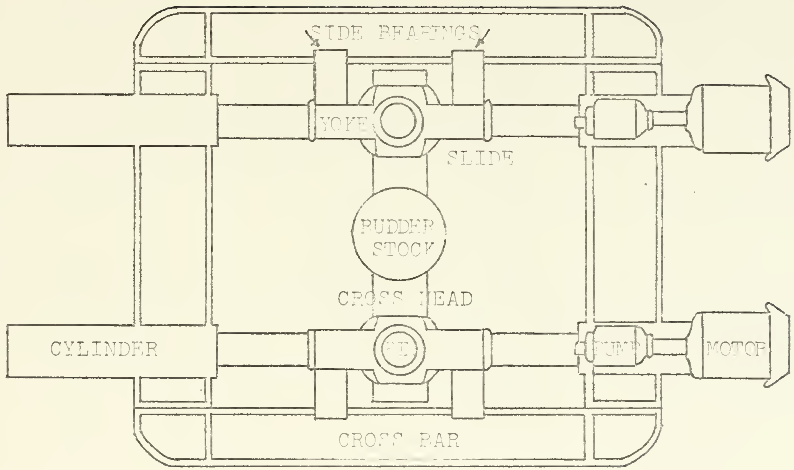



Figure 2A. Recent Steering Engine Designs.

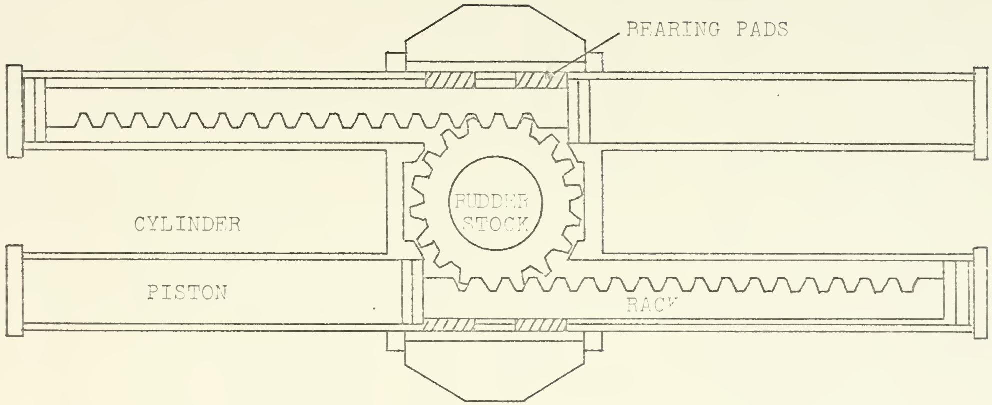



Figure 2B. Rotary Hydraulic Actuator.

It is presumed here that an adequate rudder exists and that an appropriate steering control system on the bridge provides rudder angle orders to the steering engine.

The problem is defined as one of comparing the designs of an automatic position control device with an arc of travel of 70° and an accuracy of .25° static error which uses 440 volts A.C. electrical power to produce five million inch pounds of torque on a shaft turning at .39 rpm. That is, a high torque, low speed, electrically powered, position servo.

## B. ANALYSIS AND SIMULATIONS

### 1. General

The mechanisms which will be described will be grouped into three general categories:

1) Direct acting Electromagnetic Device
2) Electro-hydraulic Machines
   a) Linear actuator or piston and cylinder driving through either a rapson slide, a tiller and linkage or a rack and pinion.
   b) Rotary vane actuator
3) Electro-mechanical machines
   a) Gear reduction drives
   b) The ball bearing screw actuator
   c) Hydrostatic bearing devices.

### 2. Direct Acting Electro-Magnetic Devices

These could take the physical form of a giant synchromotor, a D.C. motor, or an open squirrel-cage motor

all attached directly to the rudder stock. In order to get the required torque, enormous windings would have to be employed which, in addition to their great weight and size, would consume large amounts of power in resistive heating. This not only produces low efficiency but a large heat removal problem as well. And, of course, the D.C. devices would have to have their rectifiers. Of the three, the most hopeful might be the open squirrel-cage which requires no rectifier and whose radius might be extended to the point where the required electromagnetic forces are within reason.

However, the radius required would be excessively large, and even then the system would be heavier and less efficient than existing ones.

Actually this whole category could have been dismissed immediately by considering the fact that electro-magnetic devices operate most efficiently as low torque high speed machines. This is, of course, the diametric opposite of the requirements of this application. One further implication of this is that if electric motors are used, it will be to drive oil pumps, gear trains, etc. which convert its high speed, low torque to the low speed, high torque required here.

3. Electro Hydraulic Devices

a. The Linear Actuator

In view of the performance record of the electro-hydraulic piston and cylinder a feasibility study would be

redundant. However, a weight and space optimization study is a pertinent area of interest. The general procedure to be used is to attempt to optimize the system with regard to weight and then examine the effects of this optimization on space. To do this it is necessary to arbitrarily divide the problem into manageable parts to facilitate the analysis.

The linear actuator was divided into 3 components:

1. The Piston and Cylinder
2. The Pump
3. The Shaft turning device

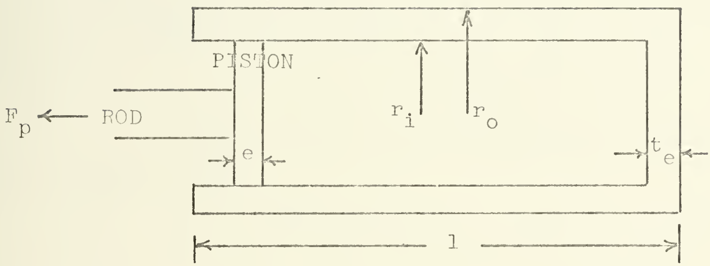
The weight optimization procedure was: first to write the equations of operations of the various components of the system. Then, the weight equations were written where possible. These equations were examined with relationship to their parameters. Then the weight optimization problem for the combined system was defined and a solution attempted.

(1) <u>Piston and Cylinder (Figure 3)</u>

The best mathematical model to use for calculating stresses in hydraulic cylinders is that of the "thick walled" cylinder. This will apply to the broadest range of cylinders, since at low pressures it will reduce to the case of the "thin walled" cylinder.

Radial stress $$\sigma_R = p_i \frac{r_i^2}{r_o^2 - r_i^2}\left(1 - \frac{r_o^2}{r^2}\right)$$

Tangential stress $$\sigma_T = p_i \frac{r_i^2}{r_o^2 - r_i^2}\left(1 + \frac{r_o^2}{r^2}\right)$$




| | | |
|---|---|---|
| $r_i$ | | Inner Radius |
| $r_o$ | | Outer Radius |
| $k$ | = | $\frac{r_o}{r_i}$ |
| $p$ | | Pressure (psi) |
| $\gamma_o$ | | Specific Weight of oil |
| $\gamma_s$ | | Specific Weight of Steel |
| $\sigma_y$ | | Yield Stress |
| $f$ | = | 1/factor of Safety |

Figure 3. Piston and Cylinder Nomenclature.

Weight equation $$\frac{Wf\sigma_y}{F_p\gamma_s\ell} = \frac{f\sigma_y}{p}\,\frac{2t_e}{\ell} + \frac{\gamma_o}{\gamma_s}\,\frac{\ell - 2t_e}{\ell} + \frac{f\sigma_y}{p}\left[\frac{3+\sqrt{\dfrac{4f^2\sigma_y^{\,2}}{p^2} - 3}}{\dfrac{f^2\sigma_y^{\,2}}{p^2} - 3}\right]$$

With these equations and calculating maximum stress, applying the Henckeg-Von Mises Shear Energy Theory the curves of optimization indicated in Figure 4 are obtained.

From the Curve the following general conclusions may be drawn:

a) Curves are flat in vicinity of minimums.

b) The minimum value decreases with decreasing $t_e/l$ but $t_e/l$ is determined by $F_p$, $f\sigma_y$, and l. That is, it is determined by application except for the value of $f\sigma_y$. $\alpha$ is a constant which depends on the supports $t_e = (\alpha F_p/\pi f\sigma_y)^{\frac{1}{2}}$.

c) $f\sigma_y$ can be chosen by taking a pressure of 5000 psi and then trying to arrive 5 or 10% higher than minimum point. Substitute this into the $t_e/l$ equation. Note that increasing $f\sigma_y$ in this way from that corresponding to the min $p/f\sigma_y$ value decreases $t_e/l$ but its effect is very small.

d) As $f\sigma_y$ goes down, the actual weight goes up proportionally because of the $f\sigma_y$ term in the non-dimensional weight coefficient. Generally $\sigma_y$ is desired as low as possible primarily for cost considerations.

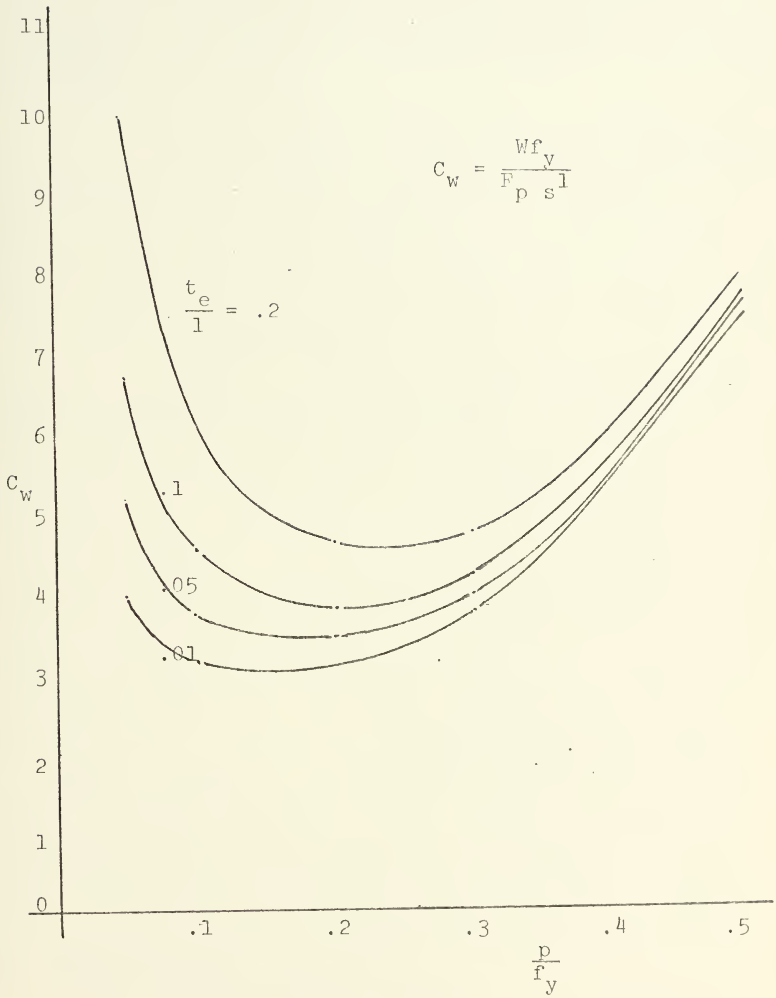



Figure 4. Optimization Curves for Hydraulic Piston and Cylinder.

e) Note the effect of increasing l is to lower $t_e/l$ but this effect is not as large as its effect on $C_w$ which causes the weight to go up proportionally with l, since

$$C_w = \frac{W f \sigma_y}{F_p \gamma_s l}$$

f) The effect of increasing $F_p$ is to cause $t_e/l$ to increase slightly and $C_w$ to increase linearly. Therefore, a proportional increase in $F_p$ will cause more increase in weight than will the same increase in length.

(2) Hydraulic Pumps

Figure 5 corresponds to the plot of weight vs. flow for hydraulic pumps corresponding to several manufacturers data. For a given pressure, weight goes up linearly with flow rate Q. The points on these lines which meet the requirements of this application can be located. Then, with some luck, an analytic approximation might be obtainable for the curve through these points. It is to be emphasized that the curve drawn does not represent the weights that would be obtained if existing equipment were used. However, an analytic expression is desired to facilitate a system weight analysis, and the curve does represent the weights to be expected if pumps were custom built for this application with the same design parameters as the existing pumps.

(3) Shaft Turning Devices

Given the same requirements of Rudder Torque, maximum piston force Fp. and the same maximum rudder angle of 35° a mathematical analysis was made to choose as

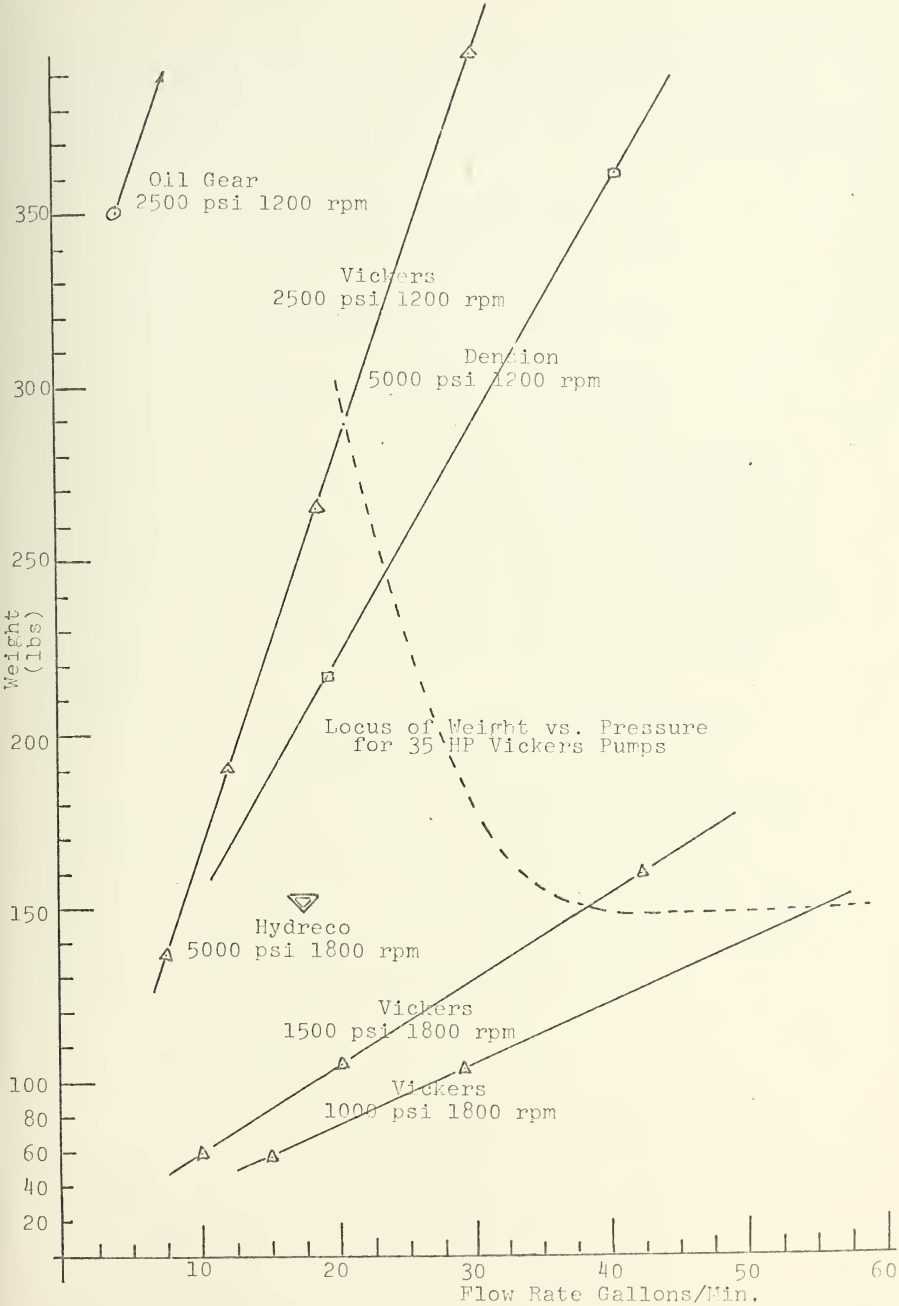



Figure 5. Hydraulic Pump Weights.

a shaft turning device one of the following elements: Tiller and Linkage, Rapson Slide, and Rack and Pinion (Figure 6).

The travel obtained was:

$$\text{Tiller and Linkage} = \frac{2T}{F_p} = .7$$

$$\text{Rapson Slide} = \frac{2T}{F_p} = .47$$

$$\text{Rack and Pinion} = \frac{2T}{F_p} = 1.221$$

The Rapson Slide, was chosen since the travel is markedly shorter for the same maximum torque T and piston force $F_p$. The cylinder length l equals the travel. It has already been noted that decreasing l will decrease cylinder weight. For this reason the rapson slide is chosen for further analysis.

(4) Optimization of the Complete Linear Actuator System

To optimize the complete Linear Actuator Electro-Hydraulic System, divide weight optimization into 3 parts: (a) crosshead and linkage group, (b) cylinder and piston, (c) pump. Consider T as given and hence constant for the problem. The weight relations are:

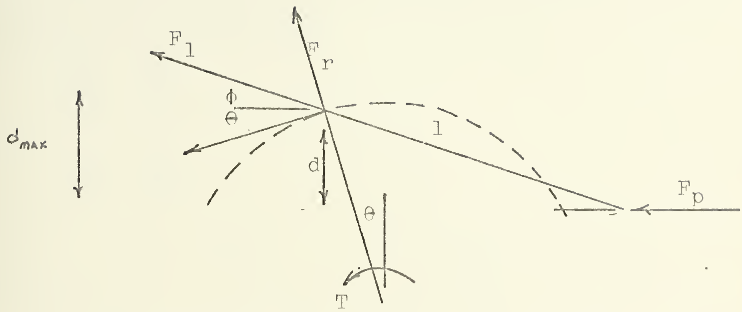
$$W_a = f(R, f\sigma_y)$$

$$W_b = f(R, p, f\sigma_y)$$

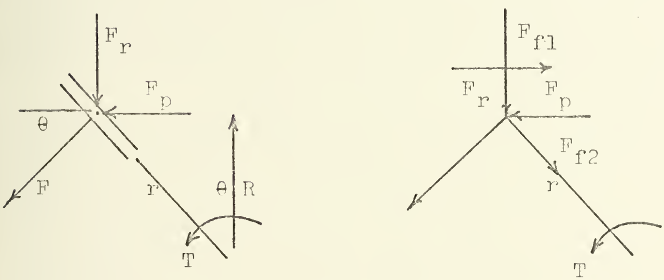
$$W = f(p, Q) \quad \text{but} \quad p \times Q = \text{Constant}$$

Therefore

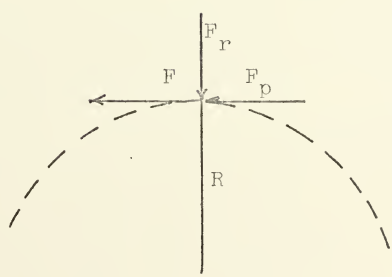
$$W_c = f(p) \quad \text{or} \quad f(Q)$$




Force Diagram for Tiller and Linkage




Force Diagram for Rapson Slide




Force Diagram for Rack and Pinion

Figure 6. Rotary Mechanism Notation

The total weight $W_t = W_a + W_b + W_c$ is a function of two independent variables R and P. $W_t\ (R,P) = W_a\ (R) + W_b(R,P) + W_c(P)$ In order to optimize, the following partial differential equations must be satisfied:

$$\frac{\partial W_b}{\partial_p} + \frac{\partial W_c}{\partial_p} = 0$$

$$\frac{\partial W_a}{\partial R} + \frac{\partial W_b}{\partial R} = 0$$

In order to solve the above equations with respect to R, it is necessary to determine the variations in weight of the rapson slide with R. The results of the calculations of the weight of several possible rapson slides for different R valves are indicated in Figure 7.

For the purposes of these calculations a design was chosen which would simplify somewhat the calculations, incorporate the most important features desired in good design and still give meaningful results (Figure 8). Inspection of this figure leads to the following observations and conclusions:

(a) Total percentage change is small (8.3%)

(b) Weight decreases irregularly with R. to a shallow minimum at about R = 40.

(c) At about R = 40, unsupported lengths of component parts become sufficiently long to cause buckling instabilities. Hence this weight is not obtainable in practice. From this point on, weights will begin to rise due to the stiffening required to prevent buckling (in addition to that shown).

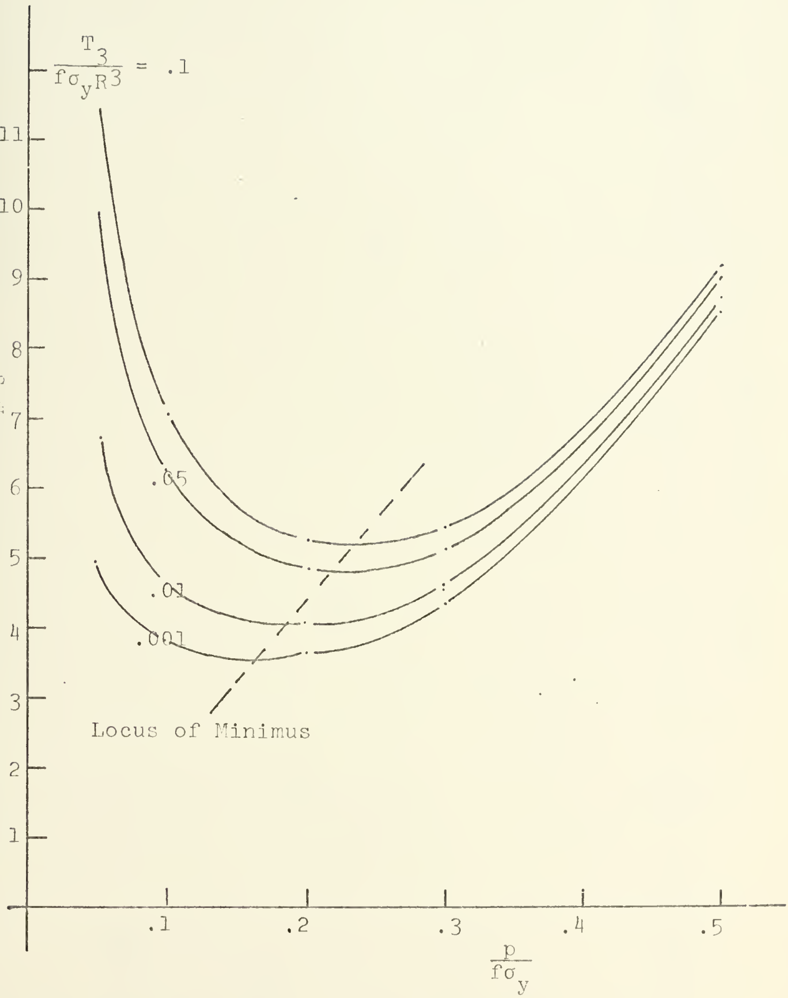



Figure 7. Optimization Curves of a Piston and Cylinder for a Rapson Slide.

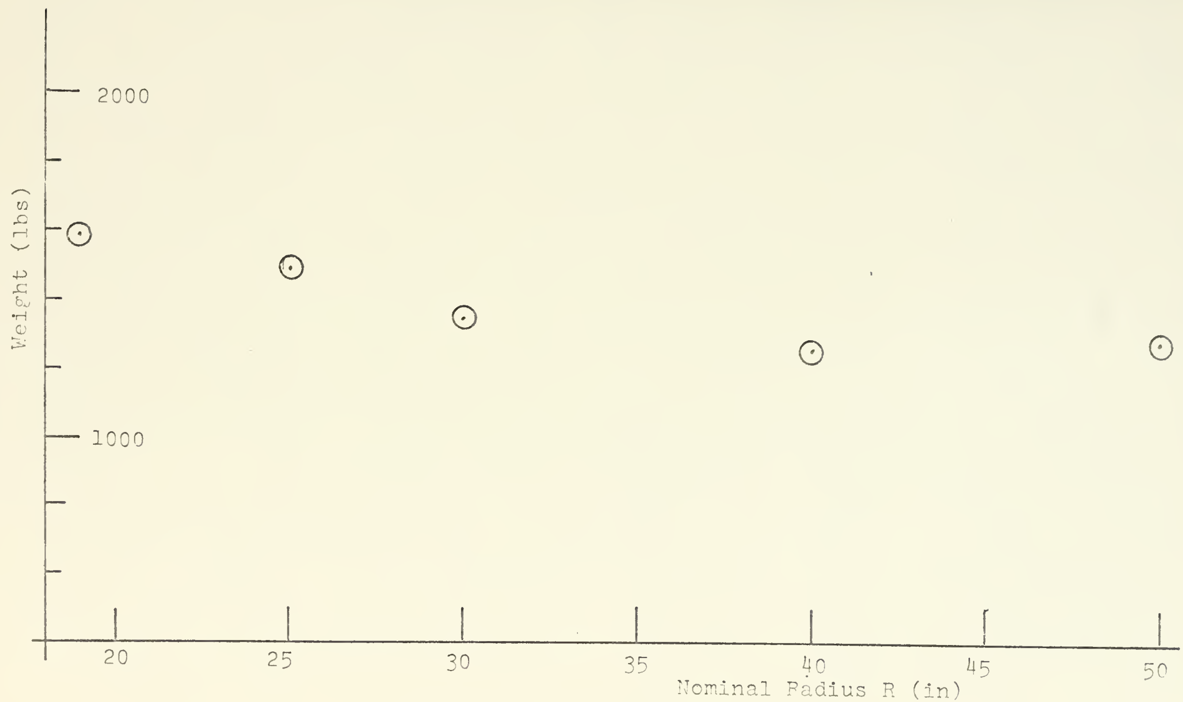



Figure 8. Rapson Slide Weights

(d) The minimum point is not clear and in general will depend on the details of the particular design.

(e) The weight of piston and cylinder decreases with increasing R (from Figure 7). Hence the optimum from a weight point of view is at that R where heavy additional stiffeners are required to prevent buckling in members.

(f) The maximum weight reduction that can be obtained in going from minimum R (most compact design) to optimum R appears to be only about 10%.

(g) Weights will decrease if the rudder stock is larger than 24". Onset of buckling occurs at a larger R. Reduction in weight by optimizing will be smaller percentagewise. However, overall weight is less.

Low weight of piston and cylinder results from the high pressure used. For instance, if the pressure is doubled, the piston area is halved, but the diameter is reduced by a factor of 4. This rapid reduction in diameter produces dramatically light weights. The higher pressure does not result in very much thicker tubes walls. This is because the thickness equation works out to be a fixed percentage of diameter for a given pressure. Therefore, if the diameter is decreased, there is a reduction in thickness to offset the higher percentage due to the higher pressure. The rapson slide is seen to be quite an efficient device resulting in only a twenty-five horsepower drive motor.

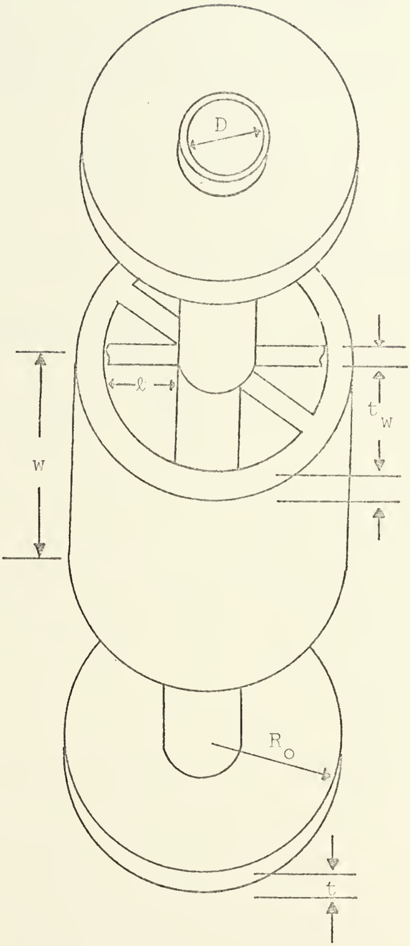
b. Rotary Vane Actuator

The rotary vane actuator steering engine has many attractive features, foremost among which is that it is the simplest mechanical device that will do the job. Along with this inherent simplicity goes ruggedness, shock resistance, compactness, and ease of installation, maintenance, and operation. Its only major defect is that it is extremely difficult to seal adequately and as a result, these units usually have high leakage rates. There is such a long length of periphery around the vanes that seals with a leakage rate per unit length acceptable in other hydraulic devices produce a total leakage rate too large in the rotary vane actuator. On the other hand, making the seals too tight in an effort to reduce the leakage, produces large friction forces which give the unit high "breakout" or no load starting torque and low efficiency under operation. Thus, it can be seen that the seals are critical parts of the rotary vane actuator design. In fact, much of the recent popularity of the device is attributable to engineering progress in this area.

Using the Hencky-Von Mises shear energy strength theory, and the nomenclature from Figure 9, the following data is found:

Shear stress at the root is $\tau_{xy} = \frac{p\ell w}{wt} = p\frac{\ell}{t}$

Stresses at the top fiber are $\sigma_x = 3p\left(\frac{\ell}{t}\right)^2 \quad \sigma_y = -p$




$$t_w = R_o - R_i$$

$$R_o = KR_i$$

$$t_w = R_i-(K-1)$$

$$\frac{D}{t} < 1.5$$

Figure 9. Rotary Vane Actuator Nomenclature.

Shaft torsional shear is $\tau = \dfrac{16T}{\pi D^3}$

$$f\sigma_y = p\left[3\left(\frac{\ell}{t}\right)^2 + 1\right] \qquad t = \frac{\ell}{\frac{1}{3}\sqrt{\frac{f\sigma_y}{p} - 1}}$$

If the maximum torsional shear is taken as $\tau_{max} = .65\ f\sigma_y$ then

$$.65\ f\sigma_y = \frac{16T}{\pi D^3}$$

This defines minimum diameter $D_{min} = \left[\dfrac{16T}{\pi(.65f\sigma_y)}\right]^{1/3}$.

$$T = \frac{-pWD^2}{4} + \frac{T^2}{pW^3}$$

$$W^4 + \frac{4T}{pD^2}W^3 - \frac{4T^2}{p^2D^2} = 0$$

The last equation defines the optimum W to minimize the periphery. An analysis of these considerations shows that although leakage rate requires careful design attention, it is not a significant factor in determining the weight and the size of rotary vane steering engine.

The great majority of the weight of the system is accounted for by the cylindrical shell and the end plates. This is because the thicknesses work out to be a given percentage of the diameter for a given pressure. Therefore, the thickness will be large for the larger diameters inherent in a rotary vane. The heavy concentration of weights in these two components suggests that attempts at weight reduction be focused primarily on the

weights. In fact, it appears that a valid initial simplification to the problem is to neglect the weights of the vanes and the oil.

Suggested ways of reducing the weight would be to elongate the unit to reduce diameter, and to use higher yield strength steel to reduce the thicknesses. The above simplifications make the problem very similar to the piston and cylinder analysis performed in the linear actuator.

The final point of interest of the design study is the relatively small size of the vanes. The span and thickness of the vanes would decrease further if the axial length of the unit were increased. These small dimensions suggest the possibility of manufacturing them as an integral part of the rudder stock by machining them directly out of the same billet.

In summary, it is clear that the rotary vane is feasible for the torque range required to drive the rudder; pressures up to 5000 psi are feasible, but their desirability must be dependent on calculations of an optimum $p/f\sigma_y$ valve.

Increasing axial length of the unit is desirable.

4. Electro Mechanical Devices.

a. Gear Reduction Drives

(1) Gear Trains. Historically, the geared quadrant has seen extensive application in steering engines. It was rugged, simple and reasonably efficient, but it has

largely disappeared from use today. In order to obtain high reduction ratios and reduce the tooth loading and the size of the drive gears, large quadrant radii were used. This made the system so large and cumbersome that it was superseded by the electro-hydraulic machines.

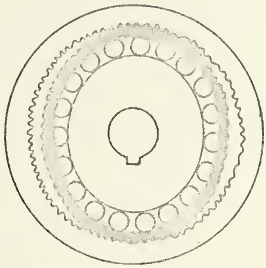
The worm gear offers a convenient lightweight design easily capable of the reduction ratios required for this application. Unfortunately, it suffers from the two defects of low efficiency and large dimensions.

Another type of gear system capable of achieving the high reduction ratios required of steering engines is the epicyclic family of gear trains. There are several sub-groups including simple epicyclic, compound epicyclic, differential, and fixed differential systems each with a myriad of possible arrangements. They may be coupled or used in conjunction with other gear types. These systems can have high capacities because the number of planet gear may be increased providing more contact area to carry the load. For high speed applications, this family of gear trains is capable of providing very high reduction ratios in an extremely compact package. However, with the high torques and low speeds involved in a steering engine, the size of the gear becomes larger and these systems lose their attractiveness. The fixed differential is capable of very high reduction ratios (up to 500:1), but of course with the same difficulties of the simple and compound epicyclic trains. In summary, an epicyclic gear train would have several

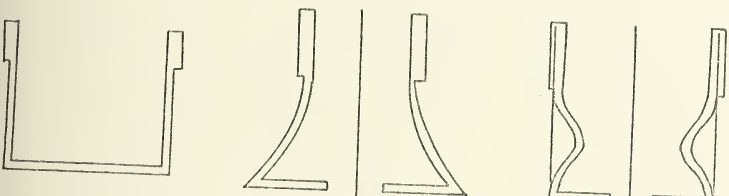
problems associated with it which could be minimized only with very extensive and clever design work. It would be complicated and have a large number of moving parts. Because of the high torque involved here, it would probably be heavier than presently existing systems.

(2) High Ratio Mechanisms. In the past, several arrangements of a screw and linkage have been used in steering engines, the Napier screw being one common example. Although this design may be made more compact than the quadrant type gear, it has an inherently low efficiency. As with the worm gear, the screw has a large amount of surface area in sliding contact under heavy load. For these reasons, its usage has generally declined.

(3) Harmonic Gear. This is a relatively new invention in which a rotating cam (Figure 10, 11) distorts a thin toothed cylinder into contact with a rigid toothed member. There is a small difference between the numbers of teeth on the flexible and rigid members so that for every rotation of the cam, these members rotate only a few tooth widths with respect to each other. The cam is called the wave guide, the flexible member the flexspline and the rigid member the circular spline. One member (any one) must be fixed and the system can be arranged so that the input and output may be taken from any member. The device is capable of reduction ratios of up to 350:1 and may be configured to meet a wide variety of applications. The spline teeth come into contact with almost pure radial




Harmonic Gear Flexspline Configuration



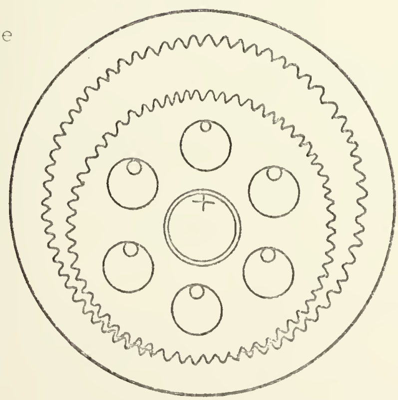
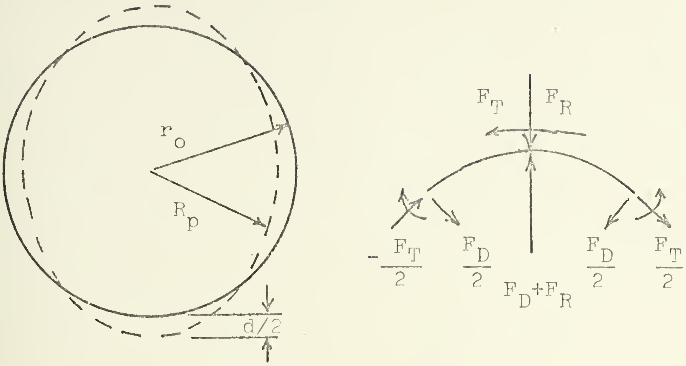
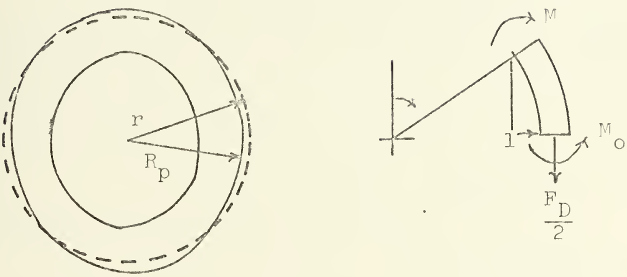




Figure 10. Harmonic Gear and Planocentric Gear

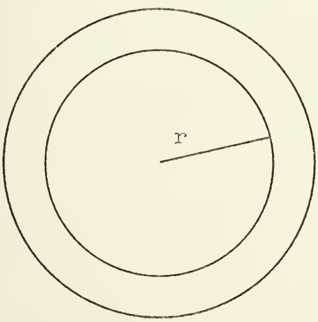



Flexspline

Wave Generator




Circular Spline




l = face width of gears, i.e., the length of the teeth in the axial direction

D = pitch diameter = diameter of flexspline

Figure 11. Nomenclature for Harmonic Gear Components

motion and therefore have extremely low sliding velocities. This results in low tooth wear and low friction losses which give the gear very high efficiencies (86% for 100:1 ratio). It has two tooth areas engaging up to 10% of the pitch diameter which gives unusually high load capacity. The components are tubular which is a particularly efficient form for torque transmission, and it allows concentric arrangement of the parts. All of these factors taken cumulatively result in making the harmonic gear the lightest and most compact of all the high reduction ratio, high torque capacity, high efficiency gear systems.

(4) Optimization of the System. When all the gear reduction drives described above are compared, the harmonic gear emerges as the most promising device, and for this reason is the only one considered.

A lumped parameter mathematical model may be constructed as shown in Figure 12.

The harmonic gear has no viscous damping, but it does have a high starting torque loss which remains about constant with time for an initial period. This is due to the requirement of a certain amount of force to deflect the flexspline, even under no load, and the ball bearing "stiction" and frictional losses at low speed. An estimate of the starting torque by the United Shoe Machinery Corporation is 350 in-lbs.

Torque losses can be represented reasonably well by a straight line plot versus load shown in Figure 12.

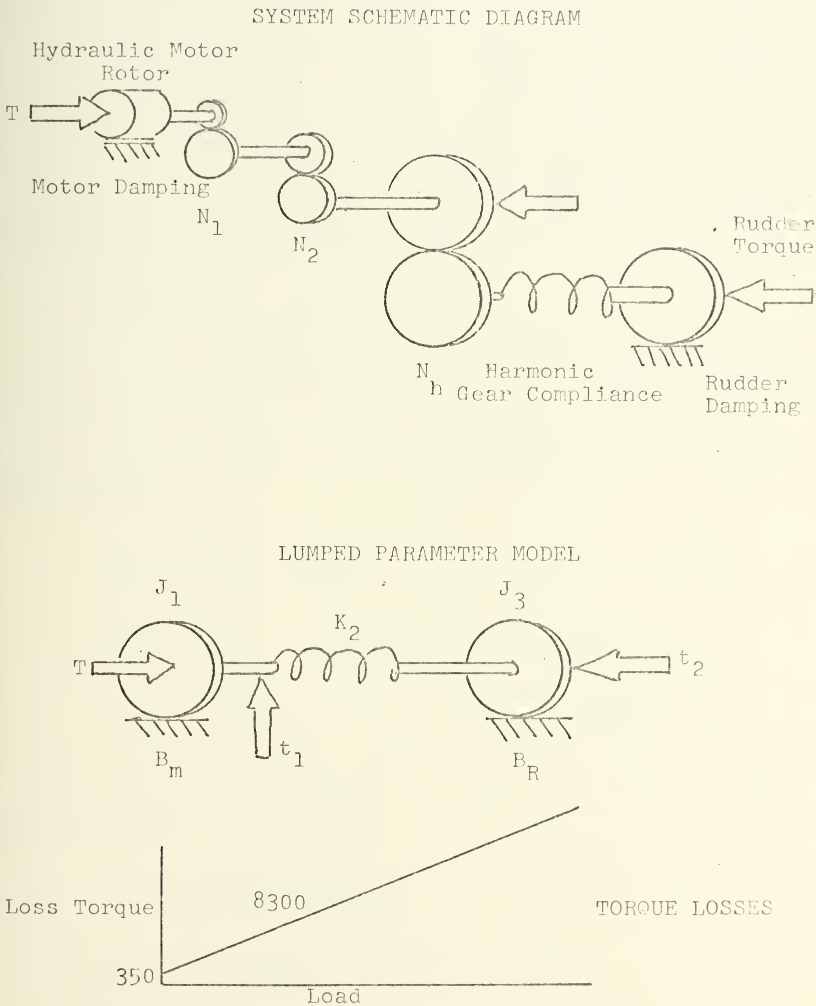



Figure 12. Mathematical Model of Harmonic Gear.

Notice that in this speed range, the losses are proportional to load and are independent of speed.

It is possible to add to this curve the estimated losses in the other gears to better represent the system.

$$T = C + K_L \theta_R$$

$$C = 350 \text{ in lbs.}$$

If the load torque is assumed to be representable as a ramp function then

$$T_L = K_R \theta_R$$

The stiffness of the harmonic gear can be estimated by use of the following formula provided by United Shoe Machinery Corporation, the firm that developed the system:

$$K_R = \frac{5 \times 10^6}{\theta} \times \frac{57.3}{\text{radian}} \qquad T_L \doteq (8.19 \times 10^6)\ \theta_R$$

$$K_H = 16000 \times D^3 \ \frac{\text{in lbs}}{\text{radius}} = \text{output stiffness}$$

$$K_H = 16000 \times (24.5)^3 = 245 \times 10^6 \ \frac{\text{in lbs}}{\text{radian}}$$

The deflection of the gear under load can be calculated now assuming $\theta = 35°$

$$\theta_\delta = \frac{5 \times 10^6 \text{ in lbs}}{246 \times 10^6 \ \frac{\text{in lbs}}{\text{radian}}} = 2.03 \times 10^{-2} = 1.165°$$

$$\text{Input stiffness} = \frac{\text{output stiffness}}{N^2}$$

The term $\frac{P\pi}{2}$ appears in all the inertia calculations and is:

$$\text{Psteel} = \frac{489.6 \text{ lbs}}{\text{ft}^3} \; \frac{\text{ft}^3}{1728 \text{ in}^3} \; \frac{\text{sec}^2}{32.2} \; \frac{\text{ft}}{12\text{in}} = 7.33 \times 10^{-4}$$

$$\frac{P\pi}{2} = 1.15 \times 10^{-3} \; \frac{\text{lb sec}^2}{\text{in}^4}$$

$$J_I = J_m(\text{motor inertia} + J_{P_1}(\text{pinion inertia})$$

$$+ \frac{J_{g_1}(\text{gear inertia})}{Mi^2} + \frac{J_{P_2}(\text{pinion inertia})}{N_1^{\,2}}$$

$$+ \frac{J_{g_2}(\text{gear inertia})}{(N_1 N_2)^2} + \frac{J_W(\text{wave guide inertia})}{(N_1 \; N_2)^2}$$

$J_W$, according to the United Shoe Machinery Corporation is:

$$J_W = \frac{W_K^{\;2}}{GN_1^{\;2}N_2^{\;2}}$$

The mathematical evaluation of the system shows that a comparatively weak spring connects a heavily damped light inertia and a large load to the drive units. This combination precludes any significant transient effects in the output. Hence, it can be seen that the springs will react primarily to the large load torque. With this in mind the system can be simplified to the model shown in Figure 13. Notice that the weak spring $K_2$ in effect isolates the heavy damping from the mechanical drive section of the system.

LUMPED PARAMETER MODEL REDUCED
BY SIMPLIFYING ASSUMPTIONS

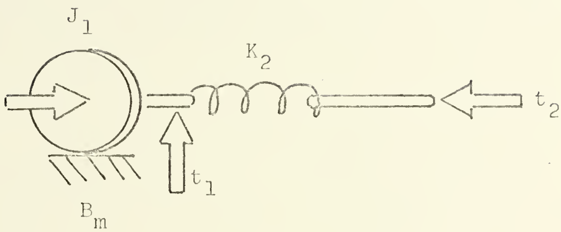



Figure 13. Simplified Model

The system equation is

$$T = J_1\ddot{\theta}_1 + B_m\dot{\theta}_1 + K_2(\theta_1 - \theta_3) + t_1$$

$$= J_1\ddot{\theta}_1 + B_m\dot{\theta}_1 + t_2 + t_1$$

If the step nonlinearity in the friction torque $t_1$ is neglected it can be written as

$$T = J_1\ddot{\theta}_1 + B_n\dot{\theta}_1 + \frac{K_R\ \theta_R}{N_1N_2N_4} + \frac{C + K_L\theta_R}{N_1N_2}$$

The damping constant $\xi$ can be calculated. It is concluded that the harmonic gear steering engine can be controlled with only position feedback when a hydraulic transmission is used.

The above model assumes that the power source (the hydraulic motor) can provide proportional control at any power level required. In actuality the motor saturates quite quickly. In order to investigate the effects of this saturation on stability, a describing function analysis is appropriate. This will determine if a limit cycle is possible under some drive conditions.

Assumptions required for describing function validity are:

(1) The system is autonomous (i.e. unforced and time invariant). $\theta_R^*$ must = 0, i.e. order rudder to amidships position.

(2) The nonlinearity is separable and time invariant.

(3) The linear transfer contains sufficient low pass filtering to warrant excluding from consideration the harmonics in the output.

The requirement of (1) which is to examine the system for possible oscillations when $\theta_R^* = 0$ is certainly logically justified because at any other $\theta_R^*$ the rudder will have a high load torque which will prevent any oscillation from occurring.

The block diagram of the system to be treated in the describing function analysis is shown in Figure 14. The saturating nonlinearity is shown with a slope of $n_1 = 1$ because the slope is already accounted for in $C_1$ and $C_2$. The analysis proceeds in accordance with Chapter 9 of J. E. Gibson's "Non Linear Automatic Control."

$$K_{eq} = g + ib = \frac{n_1}{\pi}(2\theta_2 - \sin 2\theta_2) + \frac{4M}{E}\cos\theta_2 + i(o)$$

$$\theta_2 = \sin^{-1}\frac{b}{E}, \quad \begin{aligned} n_1 &= 1 \\ b &= .244 \text{ radians} \\ M &= .244 \text{ radians} \end{aligned}$$

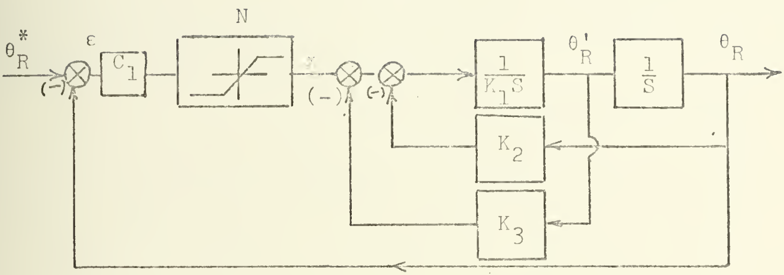
$$K_{eq} = \frac{1}{\pi}\left(2\sin^{-1}\frac{b}{E} - \sin\left(2\sin^{-1}\frac{b}{E}\right) + \frac{4M}{E}\cos\sin^{-1}\left(\frac{b}{E}\right)\right.$$

$$K_{eq} = \frac{1}{\pi}\left(2\sin^{-1}\frac{b}{E} - 2\left(\frac{b}{E}\right)\sqrt{1-\left(\frac{b}{E}\right)^2}\right. + \frac{4M}{\pi E}\sqrt{1-\left(\frac{b}{E}\right)^2}$$

$$= \frac{2}{\pi}\left[\sin^{-1}\frac{b}{E}\sqrt{1-\left(\frac{b}{E}\right)^2} + \frac{2M}{E}\sqrt{1-\left(\frac{b}{E}\right)^2}\right]$$

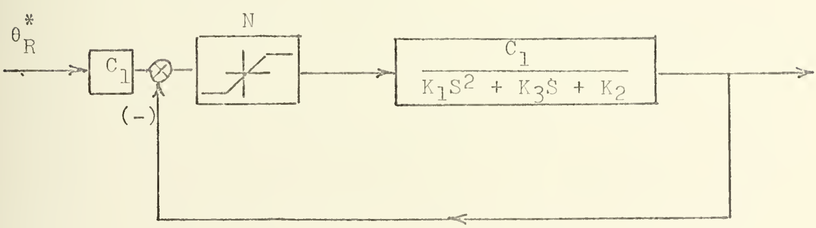
$$= \frac{2}{\pi}\left[\sin^{-1}\frac{b}{E} + \frac{2M-b}{E}\sqrt{1-\left(\frac{b}{E}\right)^2}\right]$$

but b = M because the slope = 1, so have




$K_1, K_2, K_3$ are coefficients due to a particular system

SYSTEM DIAGRAM WITH SATURATING NONLINEARITY




DESCRIBING FUNCTION DIAGRAM

Figure 14. Block Diagram for Describing Function Analysis

$$K_{eq} = \frac{2}{\pi}\left[\sin^{-1}\left(\frac{b}{E}\right) + \frac{b}{E}\sqrt{1-\left(\frac{b}{E}\right)^2}\right]$$

where E comes from letting $_R = E \sin w_t$. The condition for a sustained oscillation is

$$G = -\frac{1}{K_{eq}}$$

For a particular example, the locus of the transfer function, setting D = jw is shown in Figure 15.

It is concluded that high gain position feedback is required to meet the requirements for static error. Rate feedback is not required, if it is possible to accept some amount of oscillation in the drive end. No instabilities are indicated in the control system. Therefore, a position feedback control system used with a hydraulic transmission is a feasible method of powering and controlling the harmonic gear steering engine.

b. Ball Bearing Screw

The ball bearing screw has many advantages when used as a steering engine, the major one being that it is an efficient high reduction ratio device. It is a linear actuator and its linear motion must be converted to rotary motion by means of one of the three mechanisms discussed in connection with the hydraulic piston and cylinder. Once again the rapson slide with its low travel requires the smallest actuator. Consequently the analysis will concern itself with an arrangement in which ball bearing actuators work through a rapson slide to provide the rudder torque.

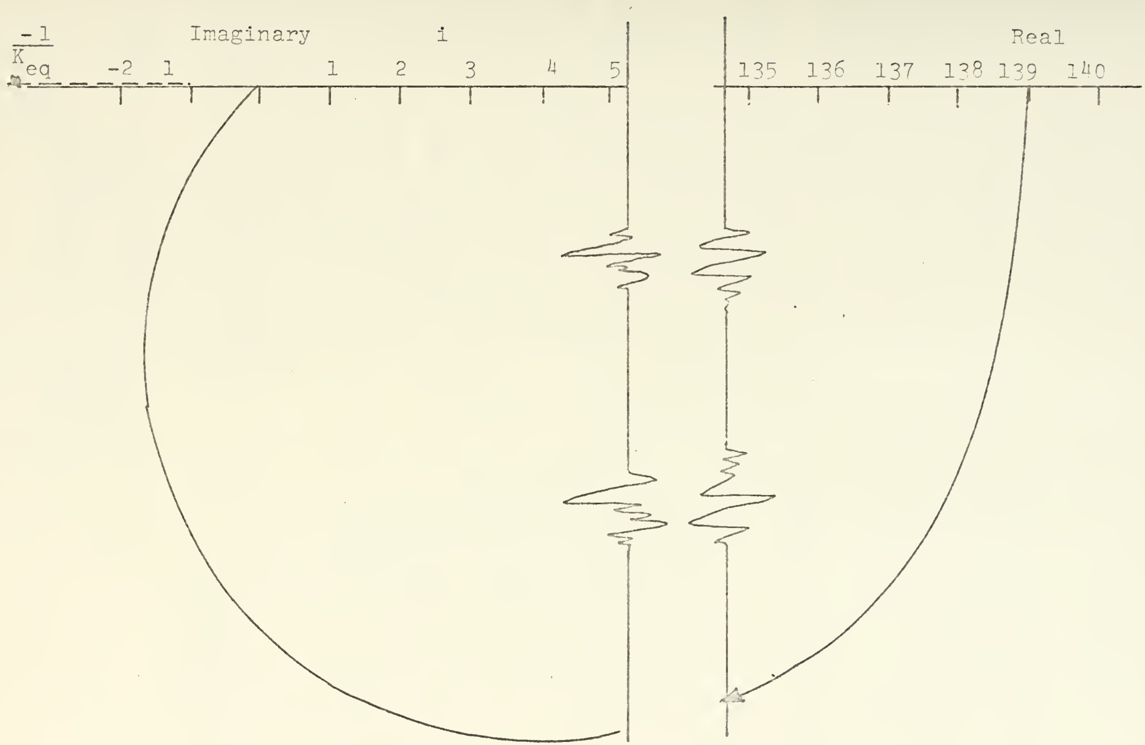



Figure 15. Plot of $\frac{-1}{K_{eq}}$ Versus G

The specific arrangement of the ball bearing screw in a steering engine considered in this analysis is shown in Figure 16.

The ball nut is located at the end and rotates with the drive gearing. The shaft does not turn and is rigidly connected to the slide block. The thrust bearing is incorporated in the bearings for the ball nut. This arrangement requires a crossbar.

The equivalent system, is shown in Figure 17A. Values of the constants are as follows:

$K_1$ equivalent spring constant of the ball bearing shaft. The shaft is loaded in either tension or compression which will cause a linear distortion. This linear distortion of the shaft is reflected in a rotational displacement of the ball nut. The ball bearing screw spring constant is the load torque divided by the angular deflection under this load. Then the ball bearing spring constant must be related back to $K_1$ in the equivalent system by dividing by the square of the reduction ratio. The calculation is as follows:

linear distortion in shaft $= e = \frac{F_p}{AE}$ ; $F_p$ includes friction forces in the slide.

angular displacement in ball nut $= \psi$

$$\psi = \frac{e\,(\text{in})}{\text{lead (in/rev)}}\, 2\pi \left(\frac{\text{radians}}{\text{rev}}\right)$$

$$\psi = \frac{2\pi e}{\lambda}$$

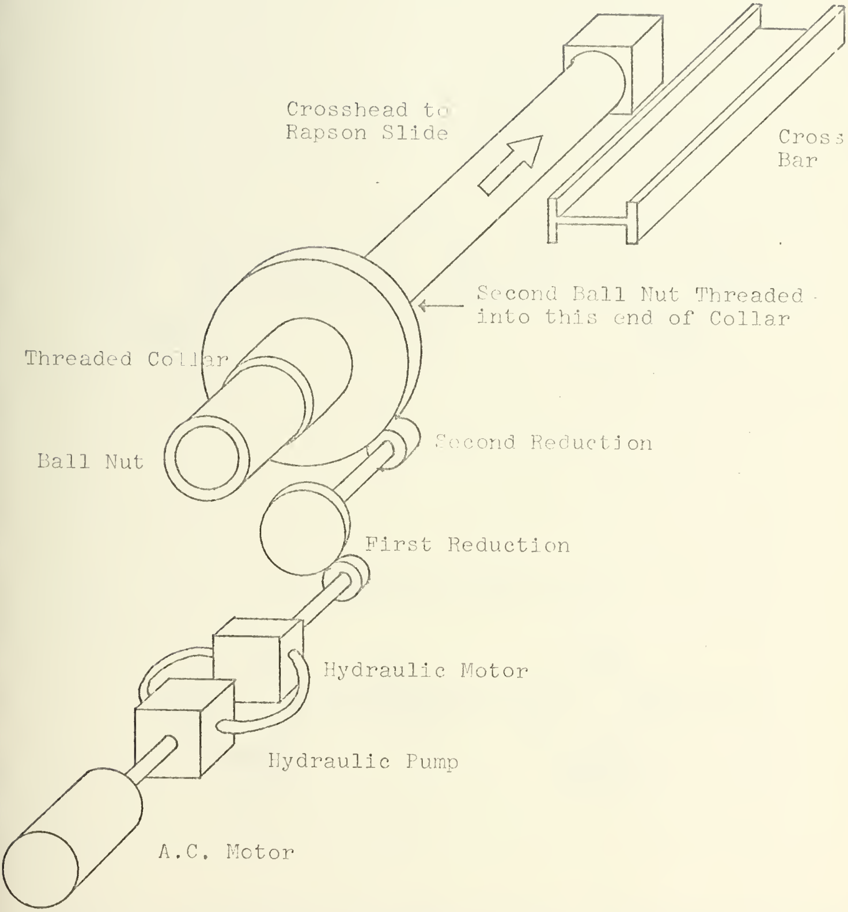
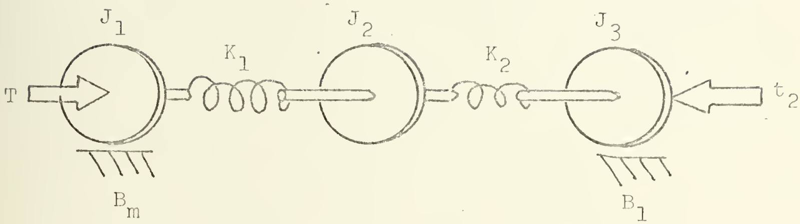



Figure 16. Ball Bearing Screw Arrangement

A. Lumped Parameter Mathematical Model




B. Simplified System Model

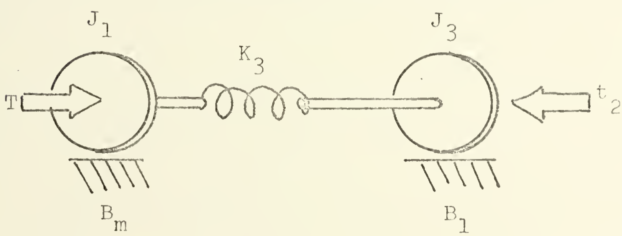



Figure 17. Mathematical Models of Ball Bearing Screw System

$$K_{BB} = \frac{T_{BB}}{\psi} = \frac{T_{BB}\lambda}{2\pi e} = \frac{T_{BB}\lambda}{2\pi} \frac{AE}{F_P \ell}$$

$$K_1 = \frac{K_{BB}}{(N_1 N_2)^2}$$

$$J_3 = \frac{J_{rudder\ \&stock} + J_{added\ mass}}{(N_1 N_2)^2}$$

Combining the spring constants into an equivalent single spring allows construction of a new model shown in Figure 17B. The equations for this model are as follows:

$$K_3 = \frac{K_1 K_2}{K_1 + K_2}$$

$$T = J_1 \ddot{\theta}_1 + B_m \dot{\theta}_1 + K_3(\theta_1 - \theta_3)$$

Writing this equation in terms of $\theta_R$, adding the hydraulic equation to the system, substituting and solving algebra the transfer function of the system may be written:

$$\frac{\theta_R}{\theta_R^*} = \frac{C_1}{AS^4 + BS^3 + CS^2 + DS + E + C_1}$$

From this equations the performance of the system is obtained. The system is heavily damped and as a result its response is that of a simple time lag. Simple position feedback has been shown to be satisfactory, and the system is stable in all modes.

c. Hydrostatic Bearing Devices

Hydrostatic bearings are a type of fluid film bearing in which the lubricant is pumped under pressure

into the clearance space between the bearing surfaces. The formation of the fluid film, therefore, does not depend on the relative speed of the bearing surfaces. Rather, it is a function of the rates of supply and leakage of the lubricant. There are several effective methods now in practice of controlling and regulating the flows in and out of the bearing depending on the requirements of the application. The analytic equations for several geometries have been solved including the radial thrust bearing. A wide variety of these bearings are available commercially. The bearing enjoys the same high efficiency under load as other fluid film bearings. But the pump which necessarily runs continuously absorbs power when there is no load on the bearing. However, with proper design, this can be kept quite small.

The concept of the hydrostatic bearing can probably be adapted to a steering engine design without too much difficulty. Then a working model would have to be constructed and tested to verify the analytic results. It is clear that the solution of this problem is well within the capabilities of present technology, and evolution of developing a practical design is certainly an avenue worthy of exploration in the future.

## C. CONCLUSIONS

Since a hydraulic piston and cylinder working through a rapson slide has the lowest overall weight and highest

efficiency it would be the best steering gear. The rapson slide has the advantage of shorter length of travel. The rotary van actuator is attractive because of its inherent simplicity, and is applicable to the full range of rudder torques. However, it is the heaviest of all the systems considered.

The harmonic gear is the most attractive of the possible gear reduction drives. The bell shaped ball bearing actuated flexspline configuration is the most suitable of these applications. The system is heavy and has the lowest efficiency of the group. The high compliance of the gear makes the natural frequency of the rudder undesirably low which introduces the possibility of rudder and hull vibrations.

The use of ball bearing screws is feasible but limited by manufacturing considerations from use in the high torque range. Ball bearing life is the critical design factor. The system is quite light in weight and has good efficiency.

Development of the hydrostatic lineary screw actuator is technically feasible and desirable.

Direct acting electro-magnetic devices are not feasible for use as steering engines.

# III. AUXILIARY MACHINERY. DISCHARGE OF BALLAST AND BILGE WATERS

## A. PROBLEM

An investigation was undertaken to provide United States naval combatant ships with the capability of avoiding oil pollution in excess of 100 parts of oil per million parts of seawater in their discharges of ballast and bilge waters. Examination of the open literature revealed that:

(1) No available technology alone, is practicable for shipboard reduction of the oil content to points below the required limit.

(2) A combination of two or more operations along with reprocessing might be suitable for the purpose.

(3) Reliable devices for measuring oil content in water are needed for satisfactory solutions to the problem.

Since no single means has been found, however, the development of a process has begun. The first goal is to solve the bilge-water, oil-pollution problem, which appears simpler, and then to extend this solution to the ballast-water, oil-pollution problem.

Naval ships have a difficult problem in observing the oil-pollution requirement. Because of the space and weight requirements aboard naval ships, any process equipment installed to treat the ballast and bilge waters before their discharge has to be compact and of minimum weight.

B. ANALYSIS

1. Separation Techniques

Separating two immiscible liquids, such as oil and water, in a ship's fuel oil tanks or bilges is not a simple process. The maintenance of two separate liquid phases is confounded by the mixing of the phases caused by the ship's motion by the action of pumps which may be present in oils (either as deliberately added performance-improving chemicals or as a result of oxidation). Thus, the oil and water inside a ballast tank or bilge can exist in forms ranging from well defined separate layers to mixture (dispersions or emulsions) of varying degrees of stability. Although some dispersions or emulsions are very difficult to separate, the bulk of one or both liquids can generally be separated by either a single technique or a combination of several techniques.

The following sections cover possible shipboard separation techniques.

a. Settling

Separating oil from water by settling should be a natural first choice. Where it can be used, it requires comparatively little energy and inexpensive equipment. In general, oil-water mextures will separate into two layers if they are allowed to stand for a sufficient period of time.

The separation of a mixture of two immiscible liquids by settling depends on:

(1) The particle size of the inner, dispersed on discontinuous, phase in the outer or continuous phase.

(2) The difference in their densities.

(3) The viscosity of the outer or continuous phase.

Three laws have been suggested for the calculation of the rate of rising (separation) of a drop of the dispersed phase, for different ranges of a characteristic property, the Reynolds number ($Re = DV\rho/\mu$). Reynolds number is an indication of flow conditions. As Reynolds number increases, flow condition changes from laminar to turbulent. The three laws used for calculating the rising velocity of a drop of the dispersed phase are:

Stokes Law (for Reynolds number below 2):

$$V_L = \frac{gD_L^{\ 2}}{18\mu_h}(\rho_h - \rho_L)$$

Intermediate Law (for Reynolds Number 2-500):

$$V_\ell = \frac{0.153\ D_\rho\ 1.14}{\rho_h\mu_h\ 0.43}\left[g\rho_h\ (\rho_h - \rho_L)\right]^{0.71}$$

Newton's Law (for Reynolds Number 500-200,000):

$$V_\ell = 1.74\left[\frac{gD_L(\rho_h - \rho_L)}{\rho_h}\right]^{0.5}$$

where

v = velocity

g = gravitational force

D = diameter

$\rho$ = density

$\mu$ = viscosity

and subscripts

h is for continuous phase

$\ell$ is for dispersed phase.

These three laws indicate that the rising velocity or separation of an oil drop is favored by larger phase-density difference, larger drop size, and lower viscosity of the continuous medium. While the density and viscosity can be affected by temperature changes, the drop size is affected by degree of agitation. The change in density and viscosity with temperature is available for oils, water and seawater. The following table serves as an example of the effect of density difference on rising velocity. In this table specific gravities of seawater, NSFO, and distillate-type fuel oil have been taken to be 1.03, 0.99, and 0.89, respectively.

EFFECT OF DENSITY DIFFERENCE ON RISING VELOCITY OF OIL IN SEAWATER

| | NSFO ($P_\ell$ = 0.99) | Distillate Fuel ($P_\ell$ = 0.89) | Increase in Rising velocity |
|---|---|---|---|
| Stokes' Law $\rho_h - \rho_\ell$ | 0.04 | 0.14 | 350 |
| Intermediate Law $(\rho_h - \rho_\ell)0.71$ | 0.101 | 0.248 | 245 |
| Newton's Law $(\rho_h - \rho_\ell)0.5$ | 0.2 | 0.374 | 187 |

Note: $(\rho_h - \rho_\ell)$ is density difference against seawater.

Separation rate increases up to 350% can be obtained depending on the nature of the liquid. Figure 18 further illustrates the relationships of drop versus rising velocity and time required for an oil drop to rise a distance of one foot in calm water. The larger the drop size, the greater the ease of separation.

Several investigators have studied the effects of hydraulic behavior and turbulence on drop size in settling. They all agree that turbulence is not a favorable condition and that rising velocity or rate of separation changes with degree of turbulence. Separation also depends on on the oil properties and residence time. The above investigators and some others also looked into separation in a flowing stream. They came to the same conclusions as did the American Petroleum Institute (API) that the faster the flow, the less favorable are the conditions for separation. The API has made recommendations for separator design as a result of their supported investigation at the University of Wisconsin. A depth width ratio of 0.3 - 0.5 is suggested. Figure 19 shows a relation based on depth/width ratio for drop size and length of separation needed for a flow rate of 3000 gpm. When sufficient time and a large area are available, continuous flow separation like that suggested is possible.

Settlers with perforated conical plates, or perforated plates of other geometry, have been built by various suppliers and used on ships for bilge water and/or

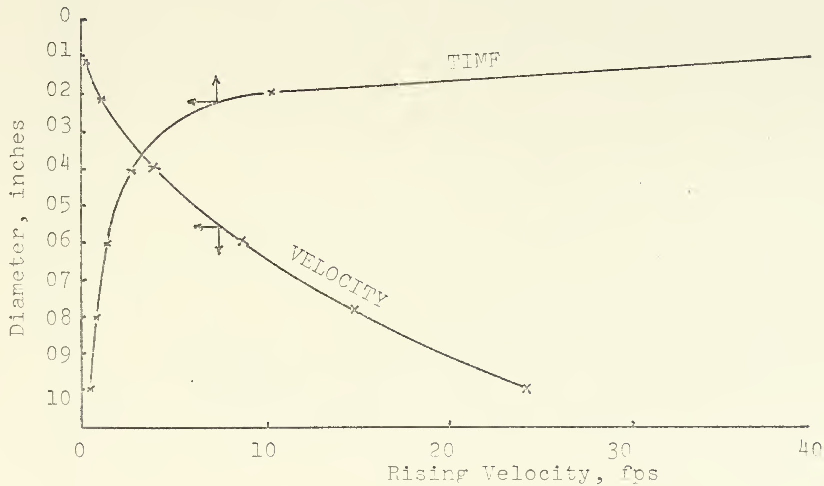



Figure 18. Rising Velocity and Time of Different Size Oil (Specific Gravity 0.99) Globules.

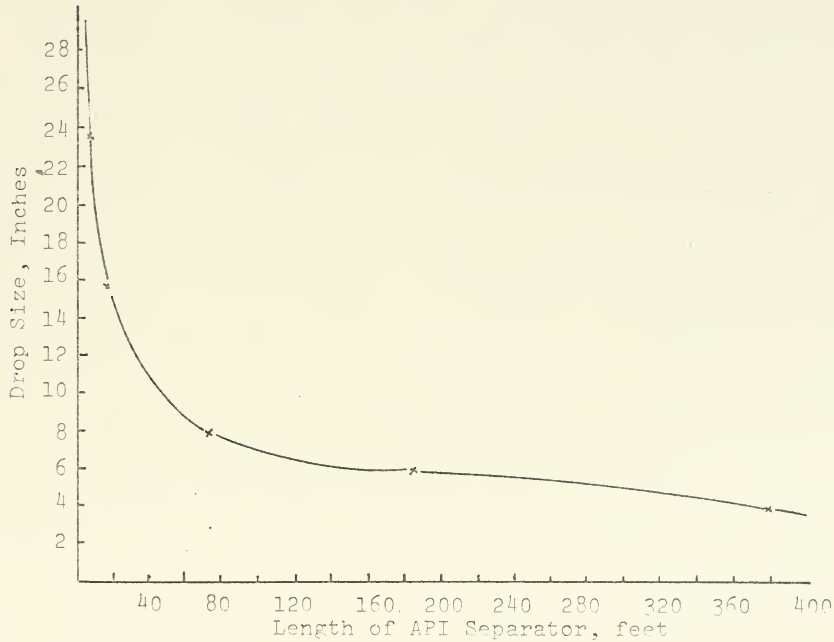



Figure 19. Diameter of Particle which will Rise to Surface of API Separator (Cross Section 40 x 7.75 ft) Versus Length Necessary for Particle to Rise to Surface (Specific Gravity 0.99, flow Rate 3000 GPM).

ballast water. They are generally suitable for low flow rates.

b. Flotation

Flotation is similar to settling in principle. It employs an essentially insoluble gas to surround and thus float suspended particles or drops by increasing the density difference between the suspended air-attached particle or drop and the continuous medium, thus facilitating the separation. The insoluble gas (usually air) may be added either in a dissolved or a dispersed form. Because of the low gas density, a drop of NSFO with an attached air bubble of the same size can rise 15.7 times faster. Even if the attached air bubble is only half its size, the NSFO drop can rise 10.8 times faster. Oil industries and public water systems have successfully applied this technique to separation problems in their settling ponds. However the ponds are usually shallow (5 - 7 feet in depth) and long (30 - 110 feet).

A word of caution seems to be in order here. Under suitable conditions certain proportions of air oil (hydrocarbon) vapor form explosive mixtures. Therefore, when such flotation technique is to be used, adequate ventilation and fire precautions must be carefully observed.

c. Centrifugation

Centrifuges have been used in removing solids from liquids or liquids from other mutually insoluble liquids. In general, when centrifuge is used for

separating one liquid from another, a minimum difference of 3% between liquid densities is recommended. This is because smaller differences would require a very large centrifugal force for separation. In theory, the same Stokes' Law used for gravitational settling applies to centrifuging but with centrifugal force utilized instead of gravitational force, Under the same centrifugal force field, a mixture of liquids will behave as follows: The liquids of higher density will have a larger momentum and move toward the outer periphery of the centrifuge while the liquid of lower density will have less momentum and move toward the inner periphery of the centrifuge. The location of the boundary between the two liquids depends on the forces on both sides of the interface. Generally, the location along the radius is related to the density as:

$$\frac{\gamma_h^2 - \gamma_i^2}{\gamma_i^2 - \gamma_\ell^2} = \frac{\rho_\ell}{\rho_h}$$

where

$\gamma_i$ = radial distance for the interface

$\gamma_h$ = radial distance for the heavy phase

$\gamma_\ell$ = radial distance for the light phase

$\rho_h$ = density of the light phase

$\rho_\ell$ = density of the heavy phase

There are two types of centrifuges used in liquid-liquid separation. Tubular-bowl centrifuges are usually of high force (in the range of 13000 g) and lower capacity (500

gallons per hour (gph)). Disk-bowl centrifuges usually can handle up to 5000 gph at a lower force (in the range of 7000 g). An equation has been derived for calculating the critical drop size $D'_\ell$, which a given centrifuge can separate. It is:

$$D'_\ell = \left[\frac{9\mu Q \quad \gamma_i - \gamma_\ell}{(\rho_h - \rho_\ell) W^2 V \gamma}\right]^{0.5}$$

where

$Q$ = volumetric feed rate of centrifuge

$\gamma$ = radius of the centrifuge

$\mu$ = viscosity

$W$ = angular velocity of centrifuge

$V$ = volume of centrifuge

The critical drop size is the drop size smaller than that which will be in the light phase and larger than that which will be in the heavy phase. This relation indicates that the critical drop size varies inversely with the rotational speed of the centrifuge, and it varies with the square root of the feed rate. Since the centrifugal force is proportional to the square of rotational speed, a decrease of half the critical drop size would require twice the speed and four times the centrifugal force. Similarly, such decrease in critical drop size would reduce the centrifugal force. Similarly, such decrease in critical drop size would reduce the centrifuge to one-fourth of its capacity. It is to be noted that when the rotational speed is increased,

the wall thickness of the bowl and weight of the centrifuge will have to be increased. If centrifuging is elected to remove oil from ballast or bilge water where large percentage of the mixture is water, it would be inefficient and uneconomical when over 90% of the liquid centrifuge is to be discarded. This does not mean that centrifuge is not a good liquid-liquid separator. On the contrary, centrifuges have been satisfactorily used in removing water from lubricating oils aboard naval ships.

d. Hydrocline

In a hydrocyclone, the liquid is forced into circular motion due to tangential injection of the liquid against the circular configuration of the hydrocyclone.

The advantage of a hydrocyclone over the centrifuge is that there are no moving parts. The initial cost is low, therefore, and maintenance is easy. It also has the advantage of being relatively light in weight. These advantages make the use of this technique of separation potentially more favorable than centrifuges on ships. It would, however, require a considerable amount of power to push the liquid mixture through at a fairly high rate in order to achieve a high-centrifugal force for separation.

Up to the present, the use of this technique in separating two immiscible liquids has not been very successful because the turbulence created by the high flow rate tends to shear and break the dispersed phase into a still finer dispersion. When the densities of the two fluid

phases are close, a separation into two phases by this method may be difficult to achieve.

e. Coalescer/Filter

Coalescence is a much studied topic but not a very well understood phenomenon. Coalescer/filter operation as applied to liquid/liquid separation essentially requires condensing smaller drops into bigger drops and then separating them according to gravity difference.

Physical interfacial film breakers used as means of coalescence have been mats, screens, and porous or fibrous woven meshes. A criterion is that it have a large ratio of surface area to volume. Usually the two liquids to be separated have different surface properties. A selection of proper material to coat the coalescing element determines which of the two liquids is to be coalesced. Generally it is the dispersed phase that needs to be coalesced.

f. Evaporation/Distillation

Separation of two immiscible liquids by alternative evaporation and condensation techniques is based on vapor pressure differences whether they are operated under normal or reduced pressures. Because of the large volume of mixture and small available space aboard a ship, these techniques are viewed as impractical. Besides, a large amount of energy in the form of steam or electricity for heating, pumping, and/or providing a reduced pressure is needed in vaporizing the water (over 90% of the mixture)

before the persistent type of oil can be removed. Although evaporation of seawater is used to provide potable water for a ship's use, persistent types of oil would still remain with the concentrated brine solution as it is discharged into the sea.

g. Freezing

Separation based on this technique utilizes the difference in the freezing points of the two immiscible liquids. This method of separation is not deemed practical for the same reasons discussed above in the case of evaporation/distillation.

h. Selective adsorption

Adsorption suggests that one of the two immiscible fluids (preferably the one of lesser quantity) is selectively adsorbed on the surface of a suitable material. The adsorbent material used to sink with the oil is found to have released oil later and is further reported to have affected marine life on the ocean bottom. Unless the oil adsorbed can be recovered or separated, ships would not only have to carry adsorbent but also the used oil-containing adsorbent. As well, this would create a storage problem on Navy vessels.

i. Unusual Physical Separation Techniques

Because of the need to produce pure and fine chemicals for industrial and biomedical use, several unusual separation techniques have been developed in various laboratories. Some of them are:

1. Chromatography

2. Sonic Separation

3. Membrane Separation

4. Electric and Magnetic Separations

j. Chemical

Certain chemicals can react with one component of a mixture and then be removed. In the case of separating oil from ballast and bilge waters, the general use of chemicals is not considered for the following reasons:

Chemicals used for separation will be in the discharged water. Over a period of time they would also become pollutants to contend with.

Experience has shown that chemicals used as demulsifying agents are very specific in their actions. No universal demulsifying agent is known.

The use of chemicals would require special materials for tanks and piping systems.

Since the amount of oil and water may vary over a wide range, trained operators or fully automated sensing and metering devices to administer the correct amount of chemicals may not be available.

k. Biological

Biological treatment for removing oil from ballast water inside a fuel oil storage tank is not favored at present. The variable conditions to be encountered in ships would make biological activity and its control difficult to maintain. Furthermore, after its use in

ballasted fuel oil tanks, the tanks would need to be thoroughly cleaned to ensure that there would be no microbes left to act upon new fuel oil charged to the tanks later. If applied to bilge water, a careful selection would have to be made for some microbes may emanate a disagreeable odor which could affect the personnel manning the nearby ship's machinery. Therefore, until better knowledge of such agents and their control is available, their use on naval ships has to be avoided.

2. Summary of Separation Techniques

Having discussed various possible separation techniques, the table on page 75 summarizes their possible adoption for ship's use.

3. Measuring Methods

Unless the characteristics and reliability of a process are fully known, means must be available to monitor and control its performance. In the present case of reducing the oil content in water to below 100 ppm (0.01%) limit, it is not known whether processes which will accomplish this objective and would not require sensing and control for their effective use can be obtained. Thus, it is necessary to inquire into the availability of methods and devices for sensing and determining the oil content in water at least to the level of 100 ppm. It is an additional requirement that the measuring methods and devices be able to function properly in a shipboard environment. A survey of measurement methods was made and is reported in the following sections according to the sensing principles involved.

SUMMARY OF POSSIBLE SEPARATION TECHNIQUES FOR SHIP'S BILGE AND BALLAST-WATER OIL-POLLUTION CONTROL

| Techniques | Potential Use Primary | Potential Use Secondary | Comment |
|---|---|---|---|
| Settling | Yes | Yes | Slow process |
| Flotation | Yes | Yes | Auxiliary process only; possibly hazardous |
| Centrifugation | No | Yes | Also suitable for removing water in oil |
| Hydrocyclone | No | Yes | Needs further development |
| Coalescer/Filter | Yes | Yes | A completely satisfactory element yet to be found |
| Evaporation/ Distillation | No | Possible | High cost |
| Freezing | No | Possible | High cost |
| Selective adsorption | No | Possible | Slow process; disposal of adsorbents needs to be considered |
| Chromatography | No | Possible | High cost & low rates |
| Sonic | No | Possible | May break emulsions or cause emulsion shattering of coalesced globules |
| Membrane | No | Yes | No satisfactory membrane yet; slow process |
| Electric/Magnetic | No | Possible | Needs development |
| Chemical | No | Occasional limited use of demulsifying agent | Requires new system equipment & trained operator; may produce another pollutant |
| Biological | No | No | May act on new fuel charged unless thoroughly cleaned out from tank; may present problems for personnel |

a. Electrical

Since water has a dielectric constant of 80 and petroleum oils have a dielectric constant of about 2 (2.5 for NSFO), a mixture of oil and water should have readings in between these two values. When a fluid of certain dielectric constant passes the electrodes of a capacitor, and impedance of a certain magnitude can be noticed on the circuit coupled to an oscillator. A change in the dielectric constant can alter the impedance; thus, the output signal can be used to indicate both the magnitude and the direction of change in the dielectric constant of the fluid passing through the capacitor.

It was concluded that the reproducibility of capacitance reading for a given concentration could be affected by changes in:

1. Salinity of the water phase
2. Water temperature
3. The physical surroundings of the capacitance probe.

b. Optical

The oil content in water may be detected on the basis of:

1. Light absorption
2. Optical density
3. Luminescence and colorimetry
4. Light scattering.

(1) Light Absorption. Three ranges of light spectrum-visible, infrared, and ultraviolet have been used

to measure the degree of contamination in a transparent or semitransparent fluid.

(a) Visible spectrum. The Warren Spring Laboratory of England and the Mitsubishi Heavy Industries of Japan have both developed optical instruments that pass light through the oily water and measure the oil concentration by the use of photocells. The reduction of the light intensity is used as a measure of oil content. The systems are said to be accurate within ±20% for 100 ppm of oil in water. As the wavelength of the light used approaches the size of the contamination, the absorption coefficient increases. Any secondary contamination capable of absorbing light can cause errors in reporting oil content in water.

(b) Infrared spectrum. This method also is based on light absorption. Wavelengths in the infrared range are used. Using spectrophotometers, refineries have successfully used this technique (2.84 - 3.50 μ) to measure oil in their effluent water to 0.1 ppm. This technique involves the extraction of oil from the water with carbon tetrachloride first, and the infrared absorption of the extract is then measured.

Infrared in the wavelengths of 8 - 14 has been used to survey organic pollution by aerial photography. It was reported that at the shorter wavelengths, thermal effects are not noticed.

(c) Ultraviolet. Absorption of ultraviolet light in the 210 μ 300 μ range has been reported to be used

as an organic matter pollution index of seawater arising from inland drainage or ship wastes. A continuous monitoring method or aromatic compounds in water has been reported. It uses a differential photometric system in its analyzer. Empirical calibrations are often required.

(2) Optical Density. Optical density or refractive index might be useful in cases where the nature of the constituents does not change. In the case of fuel oil and seawater, both the oil compositions and the sea-water concentrations could vary widely and therefore affect the readings. It is thus considered unsuitable for ship's application in oil-pollution control.

(3) Luminescence and Colorimetry. These techniques are not favored as means of measuring oil content on the basis of fluorescence or color in the ballast and bilge waters on ships. This is because luminescence and color change with the composition of the oily materials. In the case of oils encountered on naval ships, their origins as well as constituents could vary so much that successful application of these techniques appears doubtful.

(4) Light Scattering. The use of the principle of scattering and incident light beam when it passes through a medium containing particles as a means of measuring oil concentration in seawater presents difficulties. Other particular contamination present including air bubbles could produce an erroneous signal. The nonhomogeneity of expected particle sizes of oil in water could also give inaccurate readings. Development in overcoming these difficulties are under way.

c. Chemical Methods

Methods of determining oil in oily waters from boiler feed water, industrial waste water, and oil-filled waste water by chemicals methods have been reported. However, they are not deemed practical for shipboard use because of their complexity and the time required for execution.

d. Gas Liquid Chromatography

Use of gas-liquid chromatography has been reported for traces of organic matter in water. Whether its use would be confounded by the variety of oils to be encountered in ships is yet to be established. In view of the complexity of the expected signal which would require skilled interpreters, the use utility of this technique aboard ship is dubious.

4. Summary of Measuring Methods

The table on page 80 summarizes the possible measuring methods for sensing and determining the oil content in water.

C. SOLUTIONS

It is apparent that no single process can separate a mixture completely into its constituents without complicated and lengthy operations. The Naval Ship Research and Development Lab, Annapolis, has proposed the scheme indicated in Figure 20. It intends the separation of the oily water mixture into two bulk streams: one, water, and the other, oil. Each stream will permit its disposal until

SUMMARY OF MEASURING METHODS

| Methods | Potential Use | Comment |
|---|---|---|
| Electrical Capacitance using Dielectric Constant | Possible | Affected by salinity and other contaminants |
| Optical Light Absorption | | |
| Visible | Possible | Secondary contaminants interfere |
| Infrared (2.8 - 3.5μ) | Possible | Secondary contaminants interfere |
| Ultraviolet (210μ - 300μ) | Possible | Secondary contaminants interfere |
| Optical Density | Doubtful | Changes with salinity, dissolved matter, composition of oils |
| Luminescence and Colorimetry | No | Changes with nature of oil |
| Light Scattering | No | Non-homogeneity of particles and contaminants interfer |
| Gas-Liquid Chromatography | Doubtful | Too intricate for ship's use |

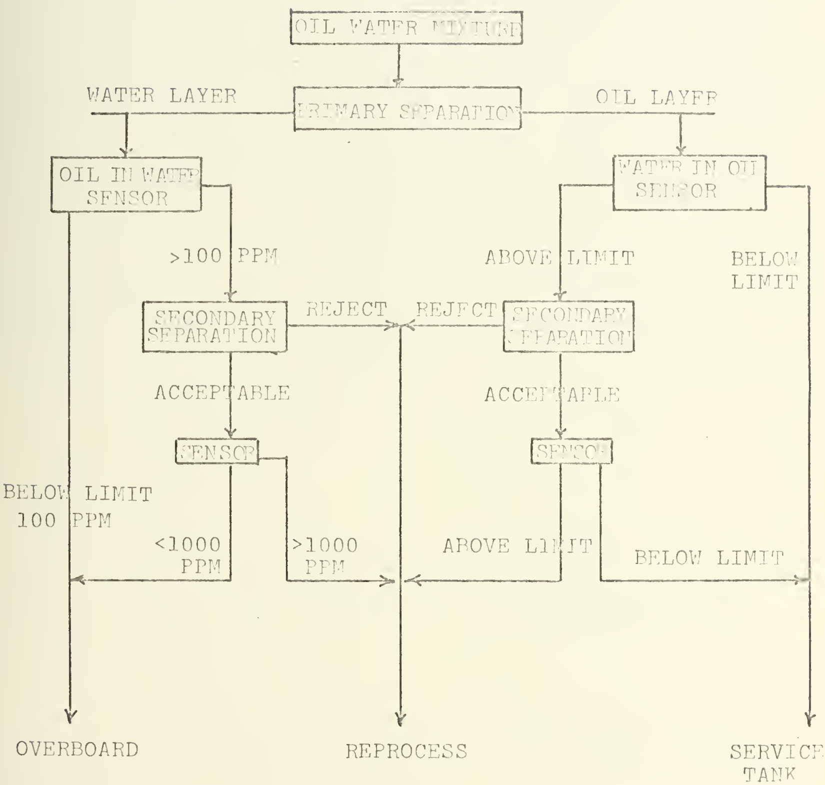



Figure 20. Scheme of Separation.

the contamination level reaches the allowed limits; the streams are then to be reprocessed.

This scheme has included the treatment of purifying the fuel oil as well. Ballasting empty fuel oil storage tanks is essential for a ship to have the needed stability and ease of maneuverability. Many naval ships have tried to avoid such operation for the fear of contaminating the fuel oil. If a ship is to be ballasted and to have control of oil pollution from its discharging water, the ship should have a satisfactory fuel purification system so that the operator may not have the worry of a contaminated oil damaging the ship's propulsion system. A total success, therefore, requires that both the water and oil purification system operate satisfactorily.

## D. CONCLUSIONS

This state-of-the-art survey indicates that, with respect to early achievement of reducing the oil content in the ship's discharging of bilge and ballast waters below the 100 ppm level:

1. A settling technique either alone or assisted by flotation, centrifugation, or coalescence/filtration techniques offers the earliest possible solution.

2. No single technique in its present state of the art is capable of completely separating the oil-water mixture to the desired level by a single operation.

3. Satisfactory measuring devices for field use still need development. Those based on capacitance or light absorption in the visible or infrared spectrum ranges are in the most advanced stages of development for shipboard use.

# IV. ROLL STABILIZATION SYSTEMS

## A. PROBLEM

The ultimate purpose of a warship is to use its weapons effectively whenever required, and this is assisted in many ways if the rolling motions are not too great. Furthermore, apart from the direct effect on the weapons system, operations such as replenishment at sea and most routine tasks are made easier if roll amplitudes are small.

A survey of the patent literature on stabilizers for roll could lead to the conclusion that such devices are innumerable, and, in the main, diverse and unrelated. The nature of the patenting process tends to create this impression. However, it is possible to classify the field of roll stabilizers in a simple and logical way.

All stabilizing systems depend on the motion of mass. They may be classified by three elementary properties: (a) the type of motion displacement or acceleration; (b) the location of the motion internal or external to the ship; and (c) the type of mass solid or fluid.

Several roll reducing systems are available but the majority of the Naval Ships use bilge keels to increase passive damping, plus moveable fins to generate a moment opposing the roll. These systems are favored as they occupy only a small volume inside the hull, but their effectiveness, especially that of the fins, is small at low forward speeds. The bilge keels reduce considerably the rolling of the

ships, but the effect on speed and the problem of retaining structural connections have limited their size.

At the same time the employment of vessels in oceanographic research puts particular emphasis on low-and zero-speed operating conditions. It is under these conditions that rolling is generally most extreme and that the stabilization of roll would be expected to show significant improvement in operational efficiency.

Historically, three types of roll stabilizers have demonstrated some significant measure of success. These are: activated fins, tanks, and gyroscopes.

As we said before, roll stabilization by activated fins depends, for its effectiveness, upon the vessel's forward speed while the other two types are potentially effective for low and zero speed conditions.

## B. ANALYSIS

### 1. Tanks - Stabilizers

#### a. General

In 1874, water chambers were installed in the upper part of a ship for the purpose of achieving stabilization against roll. The free-surface effect of these water tanks lengthened the rolling period of the ship, but also reduced the ship's stability. For this reason, the system did not find much favor. Moreover, there existed the possibility that the tanks might increase the ship's roll

instead of damping it, should synchronization occur between the roll of the ship and the flow water.

A logical development of the system was to pump the water from one leg of the U to the other, rather than rely on the ship's rolling motion exclusively for the transfer. By so doing, a limitation of the passive-tank system with its dependence on resonance could be largely overcome. The active counterpart of passive tanks was conceived by Minorsky in 1928. Minorsky reasoned that a greater degree of stabilization was possible if water were directly transferred at a high rate from one vertical leg of the U-tube to the other, in proper phase to develop a restoring moment. A full-scale installation of this system, which will be discussed later, was made on the US destroyer Hamilton. Another application of the activated-tank system was that on the World War II German cruiser, Prinz Eugen.

b. Modeling of the System Dynamics

A right-handed coordinate system oxyz, is chosen to be fixed in the ship with the origin in the ship's center of gravity (including the tank and its water). (See Figure 21.) The configuration of the tank is assumed to be roughly like a "U" tube and the tank angle $\tau$ is defined relative to the ship, as shown in Figure 22. Under the assumption of small displacements, velocities and accelerations of the ship and tank, we can set up the mathematical model and analyze each one of the components.

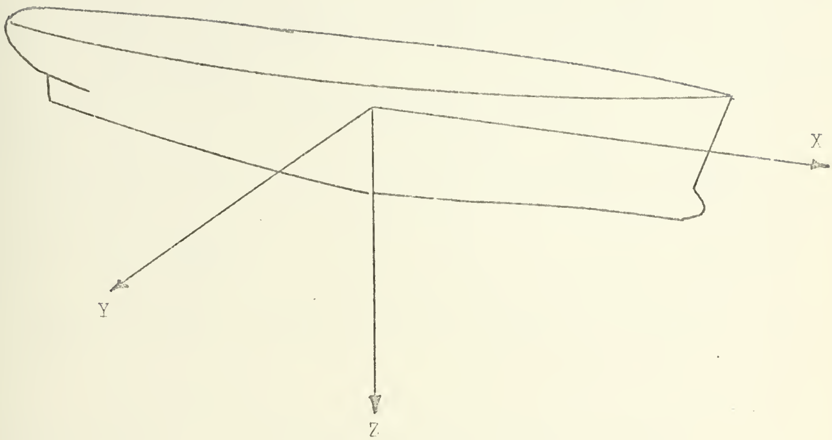



Figure 21. Ship Coordinate System.

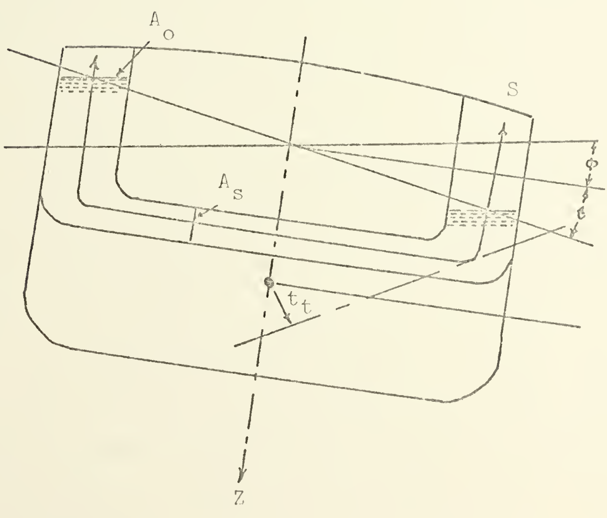



Figure 22. Tank Coordinate System.

c. Mathematical Model

(See conventions at the end of the section.)

(1) Roll:

$$[(I_x-\hat{K}_{\dot{p}})D^2-\hat{K}_p\ D-K_\phi]\phi+\rho A_o R^2[\hat{S}''D^2+2g]\tau-K_{\phi^3}\ \phi^3+Z_\gamma Y_v v$$

$$+\ \{(Z_\gamma Y_{\dot{\gamma}}-I_{xZ})D+[Z_\gamma Y_\gamma+(Z_h-Z_\gamma)mU]\}\ \gamma = K_o(t)+(Z_h-Z_\gamma)Y_o(t)$$

(2) Sway:

$$mD_v - [Y_{\dot{v}}D+Y_v](v-Z_h\rho) - [Y_{\dot{\gamma}}D+(Y_\gamma-mU)]\gamma - 2\rho A_o R^2 D^2\tau = Y_o(t)$$

(3) Yaw:

$$-[N_{\dot{v}}D+N_v](v-Z_h\rho)+[(I_Z-N_{\dot{\gamma}})D-N_\gamma]\gamma-2\rho A_o R^2 X_t D^2\tau = N_o(t)$$

(4) Tank:

$$-2\rho R(D_v+U_\gamma)+\rho R[S''D^2+2g]\phi-2\rho R X_t D_\gamma+\rho R[S'D^2+2g]\tau$$

$$+\ \tfrac{1}{2}\rho f_t R^2|D\tau|D\tau = p(t)$$

$$|\tau| \le \tau_{max}$$

d. Pump System

The pump will be an axial-flow, variable-pitch type. It is crucial to have a reliable model of its action since it is the prime activator in the control system. The conditions under which it is operating are severe. The flow is almost always changing and may, at times, flow in the direction opposite to the action of the pump. Under the assumption of quasi-steady flow, the pressure developed across the pump is a complicated function of the blade angle and inflow velocity.

It can be shown that the head across the pump is almost a linear function of blade angle, as long as the head developed by the pump is not large. If the blade angle becomes too large compared to the relative inflow angle, stall occurs and the head drops. Thus, an idealization of the complicated relation between had, blade angle, and inflow will be made. Here

$$\Delta\rho(t) = \min\left(P_\alpha \alpha_A(t),\ P_{max}\right)$$

where

$P_\alpha$ = "pump slope," pressure developed across pump per unit angle of attack

$_A(t)$ = actual angle of attack of pump blades. It is the difference between geometrical pitch angle of blades and $_i$ defined below

$P_{max}$ = maximum pressure which can be developed across pump

min( ) = implies that the minimum of listed quantities is to be selected

$\alpha_i$ = $\tan^{-1}\left(\frac{RA_o D\tau}{V_\gamma A_p}\right)$ inflow angle into pump

$A_p$ = flow area through pump

$A_o$ = the cross-section area of one of vertical legs of U-tube

$V_\gamma$ = average peripheral speed of blades

Thus,

$$p(t) = p_\alpha\left[\alpha - \tan^{-1}\left(\frac{RA_o D\tau}{V_\gamma A_\rho}\right)\right]$$

$$|p(t)| \leq P_{max}$$

It is presumed that the pump blades will be continually positioned by some servo mechanism. Because of

the inertia of the large blades, the linkages required, and so forth, it is clear that the servo motor will not be able to change blade angle immediately upon command. There will be, of course, some time required to move the blade. The exact description of the blade angle, $\alpha$ as a function of the commanded blade angle, $\alpha_c(t)$, is likely to be very complicated and is likely to depend on the exact configuration of the servo mechanism. The essential feature will be a delay or time lag between $\alpha$ and $\alpha_c$. In addition, there will be some maximum value of $\alpha$ obtainable since it is a physical system. Therefore, it is assumed that an angle limited, simple first-order system is a reasonable approximation to the performance of this servo mechanism. This is described by the following equations:

$$T_S \dot{\alpha} + \alpha = \alpha_c(t)$$

$$|\alpha| \leq \alpha_{max}$$

where $T_S$ is the first-order time lag in the blade positioning. The larger the value of $T_S$ the more sluggish the response.

e. Modeling of the Action of the Seaway on the Ship

The modeling of an environment as complicated as the one encountered by a ship traveling at a given speed and heading into a random, wind generated seaway is not an easy task. In order to render the modeling tractable, it is necessary to make several assumptions. First, it is assumed that the seaway is unidirectional and results from the

linear superposition of elementary sinusoidal waves of amplitude given by the Neumann height spectrum and of random phases with a uniform probability distribution. The local wave height $\zeta(X,Y,t)$ above the undisturbed free surface plane and referred to a stationary coordinate system OXYZ is.

$$\zeta(X,Y,t) = \int_0^\infty [A^2(w)dw]^{\frac{1}{2}} \sin[(X \cos\varepsilon + Y \sin\varepsilon)]\, k(w) + w(t) + \theta(w)]$$

Where

$k(w) = \frac{w^2}{g}$ wave number (for deep water)

$A^2(w) = \frac{52.8}{w^6} \exp\left(\frac{-2g^2}{U_w^2 w^2}\right)$

$g$ = 32.2, acceleration of gravity

$U_w$ = speed of wind creating seaway (ft/sec)

$w$ = wave frequency (referred to fixed frame of reference)

$\varepsilon$ = angle between wind direction and negative x axis

$\theta(w)$ = Random phase angle function with a uniform probability distribution over interval $(0,2\pi)$

This equation represents an ensomble of sea surfaces than can correspond to a given sea state, and wind speed $U_w$ relative to a reference frame fixed in space. But since the ship is moving with respect to this reference, its motion causes the seaway to appear different relative to the ship. Considering the ship moving at constant forward velocity U along the X axis and transforming axis now we have:

$$\zeta(X,Y,Z) = \int_0^\infty [A^2(w)dw]^{\frac{1}{2}} \sin x \left[(X \cos\varepsilon + Y \sin\varepsilon)\frac{w^2}{g}\right.$$

$$\left. + w_e t + \theta(w)\right]$$

where $\quad w_e = w\left(1 + \frac{w U \cos\varepsilon}{g}\right)$. The encounter frequency.

That is, the forward motion of the ship causes the wave frequency to be transformed to the encounter frequency.

The interaction of the seaway and the ship is, to be sure, a very complex phenomenon. Not only are the incident waves diffracted by the ship, but the motions of the ship create new waves.

At present no procedure is available to take this complicated interplay into account during the computation of the seaway forces. Therefore, for the purposes of investigation, the wave forces created by the seaway on the ship are computed according to the Froude-Kriloff hypothesis and are function of the shape, volume and length of the ship, and represents perturbing forces that must be considered inputs to the system.

f. Stability and Control Simulation

The problem of roll stabilization amounts to reducing as much as possible the mean-square roll angle given by the time solution of the set of differential equations of motion presented above when subjected to the perturbing forces.

On a ship stabilized by means of antirolling tanks, the nondimensional correcting torque, $K_T(t)$, is given by

$$\bar{K}_T(\bar{t}) = K_{st}(\hat{\eta}_{st}{}^2\tau'' + \tau)$$

Because $|\hat{\eta}_{st}{}^2|$ is usually less than 0.10 for most typical cases, the effect of the tank water inertia reflected in the first term of the equation is usually very small. The torque is produced mainly by the instantaneous lateral location of the center of gravity of the tank water (reflected in the $\tau$ term. Hence, the correcting torque is not available immediately after the actuation of the pump. In addition to the phase lag in the servo for the pump blades, the signal has to proceed through the tank dynamics before this torque is available. The tank dynamics are second order and the roll dynamics are also second order.

Thus, at frequencies near the roll resonance one can anticipate a phase lag of approximately 180 deg between the pressure developed across the pump and the resulting roll angle. At high frequencies, this phase lag increases to nearly 360 deg. The phase lag of the pump blade actuation system and the lag required for the pressure rise across the pump blades will further delay the response of the ship to a blade command. Indications are that it is possible to obtain equipment which will make this lag small at roll resonance frequencies but, one can expect a further lag of 90 deg through the servo-system at high frequencies.

It is necessary to use a signal which anticipates the roll motion in order to cancel the inherent phase lag of the ship and tank dynamics. Roll acceleration feedback provides a phase lead of 180 deg (for all frequencies) and provides sufficient anticipation to overcome the phase lags at roll resonance but this phase lead is not sufficient to prevent poor high frequency performance. Fortunately, at frequencies which are much higher than resonance, the ship responds to the sea only slightly. Although higher order derivatives, such as roll jerk ($\dddot{\phi}$), would further help in anticipating the roll motion. Such a motion is not practical to measure.

The control system will require, in addition to roll acceleration, roll velocity and roll angle terms in order to improve the roll response below the roll resonance.

The overall configuration of a control system for the pump is shown in Figure 23. Additional closed loops including the measurable quantities $\beta$, $\dot{\beta}$, $\gamma$, $\dot{\gamma}$, $\ddot{\tau}$ and $\dot{\tau}$ are also shown in this configuration. These loops will be used in an attempt to improve the performance of the roll information feedback loop. The controllers indicates in this configuration are chosen to be continuous linear controllers since design techniques for these controllers are well known and have been successful in this type of control problem.

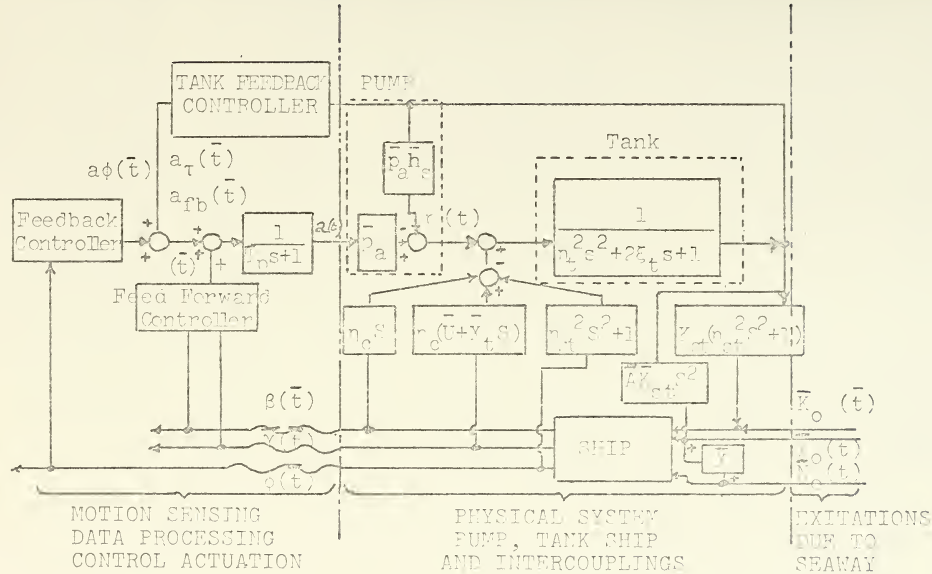



Figure 23. Roll Stabilization Block Diagram of Control Pump System

g. Results

As a result of the study of the associated computations the following facts have been established:

(1) The Design of the Tank.

(a) The tank should be designed to have as high a natural frequency as possible in order to reduce its phase lag in responding to the pump. This usually results in a tank with a natural frequency 1.2 to 1.7 times that of the ship.

(b) The tank should be designed with as large a free surface as possible consistent with the requirements of static stability. The larger the tank the smaller the required pump horsepower for a given stabilization. This usually results in a tank originated loss of GM of between 20 and 40 percent.

(c) The tank damping ratio should be determined on the basis of the control analysis treating the damping as another gain.

(2) The Design of the Pump.

(a) The blade servo should be designed to have a "cut-off" frequency of at least 1.5 times the ship's roll resonance frequency. For typical ships this leads to blade positioning system time constant of between 1 to 1.5 seconds.

(b) For an optimum control system, the mean pumping power is insignificant compared to the peak power requirements.

(3) The Control System.

(a) The major gains-roll, roll angle and roll acceleration, are best determined by a trial and error procedure such that the resultant system best damps out the motion of the ship which is subjected to an impulsive roll velocity.

(b) The optimum dimensionless gains thus chosen do not vary much from ship type to ship type.

(c) The improvement afforded by sway feed-forward is marginal and of yaw feedforward detrimental.

(d) The effect of moderate longitudinal changes in tank location is negligible.

(4) The Effect of Sea State.

(a) The roll reduction expected with an active system can be as much as 80 percent depending on the ship characteristics, sea state, wind heading, and speed. The least effect is obtained in low sea states in head seas. The greatest effect occurs in stern-quartering seas with ship at cruising speed.

(b) The effect of the nonlinearities becomes important when the water slams against the tank top relatively frequently. The major effect of this slamming is an increase of the predicted rms roll motion and a decrease in the predicted rms tank angle.

h. Conventions

| | |
|---|---|
| $Z$ | = heave perturbation displacement |
| $\theta$ and $\phi$ | = pitch and roll angles |
| $v$ and $w$ | = perturbation velocities in y and z motions |
| $p$, $g$, and $\gamma$ | = angular velocities about x, y, and z axes |
| $Y_\gamma$ and $Z_\gamma$ | = hydrodynamic (or hydrostatic) forces in y and z directions per unit motion $\gamma$ |
| $K_\gamma$, $M_\gamma$ and $N_\gamma$ | = hydrodynamic (or hydrostatic) moments about x, y, and z axes per unit motion $\gamma$ |
| $D$ | = time derivative operator |
| $R$ | = distance from ship centerline to center of a tank vertical leg |
| $ft$ | = viscous head loss coefficient for fluid in tank configuration |
| $\rho$ | = mass density of tank fluid |
| $A_o$ | = cross-section area of each vertical leg of tank |
| $S'$ | $= \int_0^L \frac{A_o dS}{A(S)}$, effective length of "U" tube |
| $S''$ | $= \int_o \frac{t_t dS}{R}$, effective coupled length of "U" tube |
| $g$ | = acceleration of gravity |
| $m$ | = mass of ship |
| $I_\xi$ | = moment of inertia of ship and tank with respect to $\xi$ axis; $\xi$ represents any axis |
| $I_{\xi\zeta}$ | = product of inertia of ship and tank with respect to $\xi$ and $\zeta$ axes, ($= \int \xi\zeta\, dm$), $\xi$ and $\zeta$ represent any two axes |
| $p(t)$ | = pressure developed across pump |
| $Z_o$, $M_o$, $Y_o$, $K_o$ and $N_o$ | = hydrodynamic forces and moments applied to ship by seaway |

$Z_h$ = positive distance from center of gravity to center of buoyancy

$K_{o\dot{p}}$ and $K_{op}$ = roll added moment of inertia and roll damping due to roll motion about center of buoyancy

$\hat{K}_{\dot{p}} = K_{o\dot{p}} + Z_h Z_\gamma Y_{\dot{v}}$

$\hat{K}_p = K_{op} + Z_h Z_\gamma Y_v$

$S'' = S'' - 2(Z_n - Z_\gamma)$

$Z_\gamma = A_n \left[\frac{m}{m - Y_{\dot{v}}}\right]$

$P_\alpha$ = "pump slope" pressure developed across pump per unit angle of attack

$\alpha_A(t)$ = actual angle of attack of pump blades. It is the difference between geometrical pitch angle of blades and $\alpha_i$ defined below.

$P_{max}$ = maximum pressure which can be developed across pump

min { } implies that the minimum of listed quantities is to be selected

$\alpha_i = \tan^{-1} \frac{RA_o D\tau}{V_\gamma A_p}$ = inflow angle into pump

$A_p$ = flow area through pump

$A_o$ = as before, the cross-section area of one of vertical legs of U-tube

$V_\gamma$ = average peripheral speed of blades

$$\alpha_e = \min\left\{ \alpha \left[ \frac{P_{max}}{P_\alpha} - \tan^{-1}\left(\frac{RA_o D\tau}{V_\gamma A_p}\right)\right]\right\}$$

2. Fin Roll Stabilization

a. General

Control systems may be classified as continuous or discontinuous, the latter case being where the fins take up extreme positive or negative incidences. For a given fin area, this results in the maximum reduction of roll but gives increased residuals of the higher derivatives of roll.

Within the capacity of the system, continuous control is preferred as it gives a smoother motion and requires less power to position the fin.

The input signal to the controller can be either feedback (Figure 24A) derived from the motion of the ship, or feedforward (Figure 24B) corresponding to the waveslope causing the roll. Since sensing the ship motion is easier than sensing the wave slope, feedback control is used.

b. Mathematical Model

As a starting point, let us consider the performance of a highly idealized system. It will be assumed that:

1. effective waveslope is the only input

2. the ship equation is linear

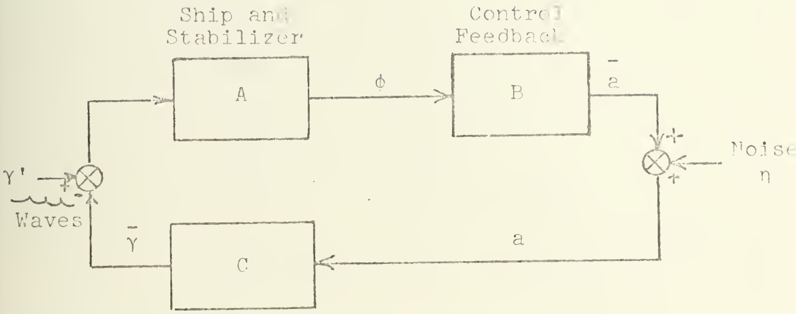
3. the control system is ideal, i.e., introduces no extraneous dynamics

4. the fin lift characteristic is linear, i.e., lift is proportional to angle of attack

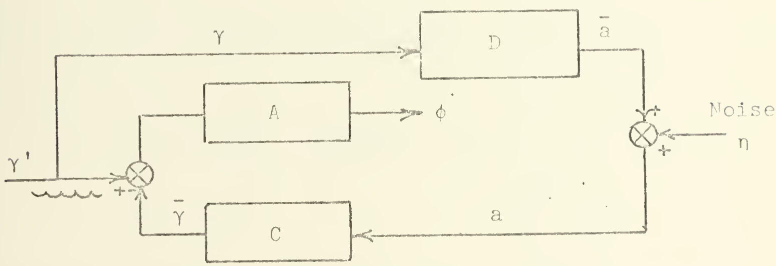
5. the control mode is feedback only, using signals proportional to roll angle, roll rate, and roll acceleration.

The block diagram is as shown in Figure 24A with the "noise," $\eta$, assumed to be zero. These assumptions lead to the following equations for the ship, control, and stabilizer:

Ship: $$\frac{\ddot{\phi}}{W_S^2} + 2\ \zeta_S \frac{\dot{\phi}}{W_S} + \phi = \varepsilon = \gamma' - \bar{\gamma}$$




A. Stabilized Ship with Feedback Control




B. Stabilized Ship with Feedhead Control

Figure 24

Control: $\alpha = K_1\phi + K_2 \frac{\dot{\phi}}{W_S} + K_3 \frac{\ddot{\phi}}{W_S^2}$

Fin: $\bar{\gamma} = K_f U^2 \alpha$

where:

$\phi$ = angle of roll of ship

$\alpha$ = angle of attack of fin

$\gamma$ = effective waveslope

$\gamma'$ = $\gamma + 2\zeta_S\gamma$ = disturbing moment

$\bar{\gamma}$ = stabilizing moment of fins, expressed in terms of equivalent waveslope

U = ship speed

$\rho$ = d/dt

and the K are constants of the control system and stabilizer. The transfer functions of the three system blocks are:

$$A = \frac{\phi}{\gamma'} = \frac{1}{(\frac{P}{W_S})^2 + 2\zeta_S(\frac{P}{W_S}) + 1}$$

$$B = \frac{\alpha}{\phi} = K_3(\frac{P^2}{W_S}) + K_2(\frac{P}{W_S}) + K_1$$

$$C = \frac{\bar{\gamma}}{\alpha} = K_f U^2$$

The overall stabilized response of the system is

$$\frac{\phi}{\gamma'} = \frac{1}{(1+K_1K_fU^2) + (2\zeta_S+K_2K_fU^2)\ \frac{P}{W_S} + (1+K_3K_fU^2)(\frac{P}{W_S})^2} = \frac{1}{A^{-1} + BC}$$

Examination of the overall stabilized response equation shows that as the control sensitivities are increased, roll will decrease. It also indicates that the

system is self-stable (in this idealized case) for any positive values of the control sensitivities, no matter how large. This explains, to a large extent, the high percentage stabilization that is possible with activated fins using feedback controls.

On further examination of this equation it is apparent that positive roll-angle control increases the apparent GM of the ship (without at the same time increasing the disturbing moment), that roll-rate control increases the apparent damping, and that roll-acceleration control increases the apparent damping, and that roll-acceleration control increases the apparent inertia. Since damping is what is needed most, roll-rate is the primary control signal for fins. Roll-angle control improves performance at the lower frequencies encountered in following seas, and roll-acceleration control improve performance against the higher frequency components met when the seas are abeam or forward of the beam.

We also see, that for given values of the control sensitivities, $K_1$, $K_2$, $K_3$, the action of the stabilizer is proportional to the square of speed. This strong relation is undesirable and should be compensated for in any system that is required to work for an appreciable range of speeds. Compensation can be achieved by adjusting the control sensitivities with speed, according to an inverse square law.

Only the inputs due to effective waveslope itself were considered in the idealized case just discussed. It is also necessary, of course, to consider rolling moment due to the drift angle, β. It would appear, offhand, that these two inputs should have the same stabilization dynamics, since they constitute indistinguishable moments in the roll equation. This will be the case where feedback control only is employed. Where a combination control of the apparent vertical type is used, the stabilization dynamics for the two inputs will not be the same. This is because effective waveslope and drift angle enter in a different way into transverse acceleration, which is detected by an apparent vertical control. Thus we must add to our inputs not only the drift-angle input into roll but also the effective waveslope and drift-angle inputs into transverse acceleration. The following equations demonstrate this point:

Equation for the rolling of a ship in waves:

$$\frac{\ddot{\phi}}{W_S^{\,2}} + 2\zeta_S \frac{\dot{\phi}}{W_S} + \phi = 2\zeta_S \frac{\dot{\gamma}}{W_S} + \gamma - \lambda_\beta (K_\beta U^2)\beta$$

where

$\phi$ = angle of roll, radians

$\beta$ = angle of drift, radians

$\gamma$ = effective waveslope, radians

$W_S$ = natural frequency of ship

$\zeta$ = damping ratio

$\lambda_\beta$ = normalized "yaw-heel" coupling coefficient

Effective waveslope and drift angle have the opposite sign in the roll equation and the same sign in the transverse acceleration equation. Physically, this means that under the effect of waves the ship tends to roll to the apparent vertical, while under the effect of drift angles it tends to roll in a sense opposite to the motion of the apparent vertical; i.e., it banks out instead of in. As it happens, all this is quite compatible with the working out of a practical apparent-vertical control.

There is one further category of inputs that is of considerable importance, particularly with regard to the heavy-weather performance of the system; i.e., the so-called false angle of attack inputs.

These false angles of attack arise out of the fact that roll, pitch, heave, and wave motion disturb the local streamlines in the vicinity of the various fins, and thereby cause their actual working angles of attack to be different from their angles of tilt versus the hull. Obviously, these disturbances are of the type described by $\eta$.

In light to moderate weather these angle-of attack disturbances are of little or no importance. However, in heavy weather they are quite appreciable and detract from the performance of the system at a time when there is little performance to spare. Over and above the matter of stabilization performance, these false angles, if not compensated, can overload the fins, and may in this manner have been partially responsible for the fatigue failures.

Calculations indicate that false angles due to either pitch or heave can easily reach 5 deg, while false angles due to wave motion may be expected to exceed 15 deg in heavy weather. Obviously, the addition of these angles to the already existing ordered angles can drastically modify the behavior of the fins. These angle-of-attack disturbances are not necessarily the same at each fin so that there are as many inputs of this kind as there are fins. Let us restrict ourselves, for the sake of the argument, to the usual case of two fins, one per side. In this case it is convenient to define two new inputs, linearly related to the actual angle-of-attack inputs at the two fins. These will be termed the "symmetrical angle of attack input" and the "anti-symmetrical angel of attack input." The symmetrical input acts equally on each fin but in the opposite sense, so that on both it acts either with or against the ordered angle of attack. The anti-symmetrical input acts equally on each fin in the same sense, so that it acts on one with and on the other against the ordered angle of attack.

Algebraically

$$\eta_1 = \eta_S + \eta_{a-s}$$

and

$$\eta_2 = \eta_S - \eta_{a-s}$$

where

$\eta_1$ = false angle of attack on starboard fin

$\eta_2$ = false angle of attack on port fin

$\eta_s$ = symmetrical false angle input

$\eta_{a-s}$ = anti-symmetrical false-angle input

No generality has been lost here, since any actual false angles can be created by the proper choice of symmetrical false-angle inputs.

These newly defined inputs have a direct physical significance. The symmetrical angle input, since it acts oppositely on the two fins, tends to create a net rolling moment. Hence, it will be detected by the stabilization system and at least partially eliminated. The anti-symmetrical angle input, since it acts in the same sense on the two fins, will not be felt appreciably until one or the other fin overloads and commences to stall. At this point the system will call for more stabilization effect, throwing the overloaded fin even further into the stalled condition. By its very nature, this last effect cannot be detected by the ship until it is too late. It can only be eliminated by curative measures applied at the individual fins.

c. Control Design Considerations (Figures 25, 26)

As was brought out in the discussion of the idealized case, three signals are needed for feedback control; namely, roll angle, roll rate, and roll acceleration. Generally speaking, this calls for three corresponding

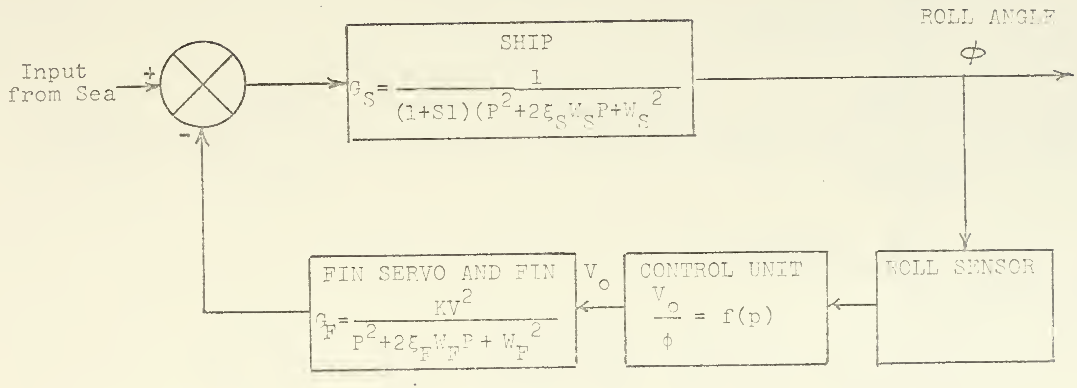
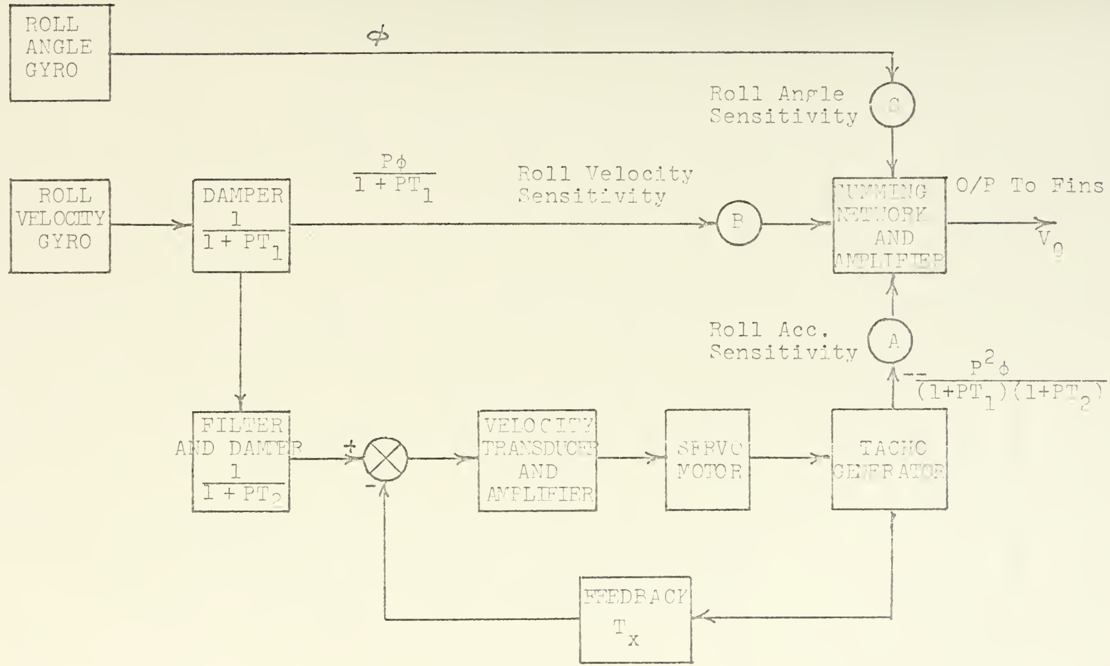



Figure 25. Ship and Control System Block Schematic
Fin Roll Stabilization




$$\text{Transfer Function } \frac{V_o}{\phi} = a\frac{AP^2}{(1+PT_1)(1+PT_2)} + \frac{BP}{(1+PT_1)} + C$$

Figure 26. Fin Roll Stabilization. Basic Block Diagram of Existing Control System

sensing elements, although the roll-acceleration signal, for example, could be obtained by taking the derivative of roll rate. In any event, the sensing elements used must meet rather high requirements on sensitivity, resolution, and accuracy. They should also introduce a negligible amount of lag into the system response. Finally, they should have the ruggedness and very long life expectancy required in marine applications.

Roll rate is, of course, the primary control signal and should be measured directly, not derived. This signal usually will be obtained from a conventional gyro. Where a feedback control only is used, the roll signal is obtained from a true vertical reference such as a gyro-vertical. Where a combination control of the apparent vertical type is used, the roll-angle signal can be obtained from a linear accelerometer mounted so as to be sensitive in the sway axis. Besides sensing roll (from the cross-deck component of the gravity vector), this instrument will pick up transverse accelerations and hance signals proportional to effective waveslope and drift angle. The effective waveslope signal provides a negative feedahead action which creates the desired apparent vertical stabilization in waves; the drift-angle signal provides a positive feedahead action which results in particularly effective stabilization against yaw-heel effects. True vertical control is probably to be preferred in military applications, but where passengers or cargo are involved, apparent vertical

control may be more suitable and slightly more economical as well.

The roll-acceleration signal is not absolutely necessary, and, as a matter of fact, its effect will hardly be apparent in the roll-angle record. It will help to control the higher derivatives of roll and so provides a smoother ride when the disturbance frequencies are somewhat above the natural frequency of the ship. This signal also tends to compensate response lags in the positioning servos. The roll-acceleration signal may be obtained by direct measurement with an angular accelerometer or by taking the derivative of roll rate. It might be noted that the roll-acceleration signal responds instantly to torques applied to the ship, and hence is particularly helpful in eliminating the effect of symmetrical angle-of-attack disturbances, where these are not eliminated at the fins. It will, of course, be of no use against anti-symmetrical angle disturbances.

The control amplifier has several functions. By unloading the sensing elements it improves their accuracy and prevents interaction effects. It also mixes the control signals and provides for sensitivity adjustments. Further, it may be used to provide a limit on ordered fin angle or ordered on hit, as the case may be. While several types of control amplifiers may be employed, the inherent sensitivity of an electronic control amplifier is of considerable advantage in the performance of these functions.

The output of the control amplifier is proportional, within capacity limits, to the required stationary moment. It is the function of the positioning servos to tilt the fins in such a manner that the required stabilizing moment is actually obtained. The fins are, in effect, horizontal rudders and the positioning torques they require are comparable to those that have to be dealt with in steering. However, the angular rates required for fins are several times greater than those required for rudders. This calls for high-power, high-performance positioning systems. Hydraulic drives, especially of the variable-displacement type, are well suited to this application. It is clear that if shock and vibration are to be avoided the whole drive system must be particularly rigid and free of backlash. There remains the problem of dealing with the false angles of attack. They can, of course, be ignored, in which case some stalling and overloading of the fins must be accepted in heavy weather. One possible remedy would be to measure the false angle in the stream near each fin and apply this as an additional tilting signal at the individual fin. This might be called "angle of attack control" and amounts to a feedahead elimination of the false angle disturbance. A more practical and direct approach is to measure the actual lift (and hence stabilizing moment) exerted by the fin and repeat this back against the order from the control amplifier. The fin lift can be determined in several ways, the most obvious

being by measurement of strain in the supporting stub shaft. This so-called "lift control" eliminates the false angles of attack by a feedback process, the positioning servos tilting each fin until its actual lift corresponds to the ordered lift signal from the control amplifier.

There are several advantages to this latter type of control. In the first place there is a direct improvement of performance by the elimination of a disturbance at the source. A more important advantage is the fact that a positive limit can now be put on fin lift by limiting the signal from the control amplifier. This protects the fin against overload and stalling. In addition, by loading the two fins equally and preventing overloads, we will obtain the maximum system capacity and work at the most advantageous lift/drag ratios. In heavy weather, lift control should increase the effective capacity of a given fin system by at least 20 per cent, and at the same time effect an appreciable saving in stabilization power. Finally, while the fins must be moved rapidly to follow the false-angle disturbances, their motions will be smoother and less subject to shock than then the false angles are ignored and stalling occurs.

d. Fin Design Considerations

At this date, the design of stabilizing fins represents a more-or-less conventional hydrofoil design problem.

The principal requirements for stabilizing fins are symmetry with respect to positive and negative angles of tilt; high maximum lift; low drag; resistance to cavitation (for the higher speed applications); low positioning moment; great strength; and retractibility (in the usual case). These requirements, combined with certain inherent limitations, pretty well dictate the plan-form and section that the fins must have. It follows that in plan-form they must be fairly stubby, with geometric aspect ratios not much greater than 2. The tips can be square. There are some advantages to taper, but it is not essential. In section form the fins must be quite thick, with thickness ratios up to 20 per cent or more at the root. Profiles with some uniform-pressure tendencies should be used.

The evolution of stabilizing fins has shown a tendency towards higher and higher maximum lift coefficients, high lift being very desirable for most efficient operation. This trend has reached a plateau, for the moment, in the so-called flap fin, which uses a plain trailing-edge flap to augment the maximum lift coefficient of the fin.

Unfortunately recent studies indicate that the performance of the flap fin is not as satisfactory when subjected to angle-of-attack disturbances as it is when being operated in an undisturbed stream.

There are two important drawbacks associated with operation in a disturbed stream. In the first place, loads resulting from fin tilt are applied approximately

0.35 chord length aft of the leading edge, whereas loads due to false angles of attack are applied approximately 0.25 chord length aft of the leading edge. Thus there is no longer any unique hydrodynamic center, and no possibility of balancing the fin for both types of loads. In practice, it turns out that the best that can be done is to balance for fin tilt (at the 35 per cent chord) and accept the loads due to false angles of attack.

As a second drawback, it was found that the maximum lift coefficient of the flap fin is not a constant but is a function of the false angle of attack.

This maximum-lift-coefficient effect is particularly adverse when angle disturbances are not eliminated at the fin, since the same false angle that adds load also reduces the maximum lift coefficient. If nothing further were done, a system employing lift control also would be adversely affected. However, there are several rather simple ways in which the limit on ordered lift may be caused to track the variations in maximum lift coefficient, thus causing the system to use at any instant just that capacity available to it and no more, whether it be larger or smaller than the average capacity of the fin.

3. Gyroscopic Stabilization

a. General

A gyroscopic ship stabilizer consists basically of a large rotating gyro wheel with a single degree of freedom about an athwartships axis, as shown in Figure 27.

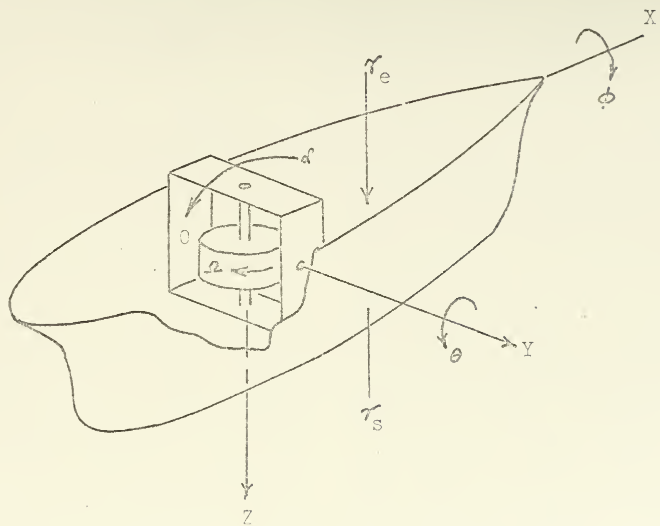



Figure 27. Gyroscopic Stabilization. Stabilizer Notation

Referring to that figure, with the gyro wheel rotating, if the spin axis Z is caused to precess (i.e.,to rotate) about the horizontal axis Y, there will result a moment in the plane defined by Z and Y. Conversely, if a moment is applied in this plane to the spin axis, the spin axis will precess.

To understand the action of this device as a stabilizer, consider that a moment in roll is applied to the ship by the action of waves, for example. With the gyro assembly free to precess about the horizontal axis, this applied moment would cause such precession. The precession velocity would give rise to a moment such as to completely cancel the effect of the roll moment upon the ship (Figure 28).

The foregoing would hold true provided that the angular momentum of the wheel were sufficiently great and the applied moment did not persist too long before reversing direction. Under these conditions the assembly would not precess too far from the initial vertical condition of the spin axis. However, the effectiveness of the stabilizer (in countering roll moments impressed upon the ship) varies as the cosine of the angle of the spin axis from the vertical; hence, with the spin axis in a horizontal position, no stabilizing action would occur. Since a condition would be reached sooner with a wheel of lower angular momentum or later with a wheel of any larger angular momentum provided that the applied moment persisted long enough.

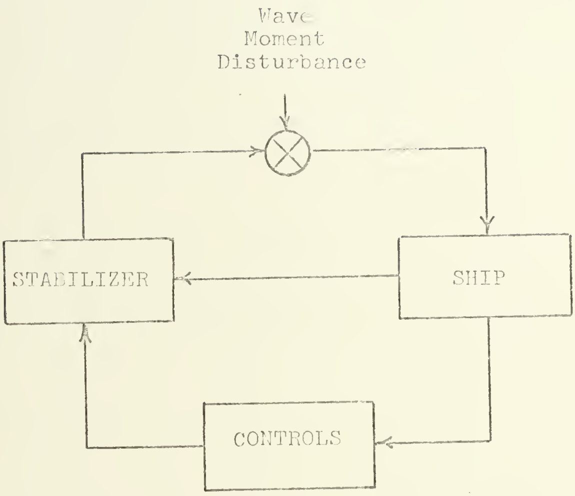



Figure 28. Simplified Block Diagram of Gyroscopically Stabilized Ship

b. Mathematical Model

(1) Ship Equation. A ship is basically an underdamped, resonant system. The equation of roll motion is assumed to be a second order linear ordinary differential equation. Various coupling effects become unimportant at low speeds since they vary as the square of speed and will be neglected. The roll equation can therefore be expressed as follows:

$$J_\phi \ddot{\phi} + B_\phi \dot{\phi} + K_\phi \phi = K_\phi(\gamma_e - \gamma_S)$$

where:

$\phi$ = roll angle, radians

$J_\phi$ = total moment of inertia of ship about axis of roll, lb-ft-sec

$B_\phi$ = hydrodynamic damping coefficient, lb-ft-sec

$K_\phi$ = static righting moment coefficient

$\gamma_e$ = effective waveslope disturbance, radians

$\gamma_S$ = effective stabilizer waveslope, radians

By definition, effective waveslope is either a disturbing or a stabilizing moment about the roll axis, expressed in terms of the static list which this moment will produce.

(2) Stabilizer Equation. The action of the gyro as a stabilizing device is obtained by controlling the velocity of precession about the athwartships axis. The response of the stabilizer to an ordered precession rate is assumed to be equivalent to the response of an ideal gyro. With the spin axis vertical, the reaction moment = $\dot{a}\Omega\frac{I}{g}$ lb-ft = $\dot{a}$ H ton-ft

where

$\dot{a}$ = precession velocity, rad/sec

I = rotor inertia about the spin axis, $lb\text{-}ft^2$

g = acceleration due to gravity, $ft/sec^2$

$H = \frac{\Omega I}{2240g}$ = rotor angular momentum, ton-ft-sec (generally fixed for a given installation)

$\Omega$ = gyro spin velocity, rad/sec, is positive in the convention of yaw, i.e., clockwise looking in the positive direction of the Z axis

a = precession angle, angle of spin axis with the vertical, positive in the same sense as pitch angle,

$\gamma_e$ is positive in the same sense as roll angle, $\phi$

$\gamma_s$ is positive in the opposite sense of $\phi$

The component of the reaction moment serving to stabilize the vessel in roll is proportional to the cosine of a. Thus the stabilizing moment = $\dot{a}H \cos a$ (ton-ft). The stabilizing moment, in terms of the static list which it would produce, is the equivalent stabilizing waveslope, $\gamma_s$. Thus

$$\gamma_s = \frac{\dot{a}H \cos a}{\overline{GM}\ \Delta}$$

where:

$\overline{GM}$ = metacentric height, feet

$\Delta$ = ship's displacement, long tons

c. Roll-Precession Coupling

As discussed above, the rolling of the vessel results in a torque serving to accelerate the gyro precession. This torque develops in the same manner as that of the stabilizing moment except that the roll velocity, $\dot{\phi}$,

operates in place of the precession velocity, $\dot{a}$. Thus, the precession torque due to roll velocity = $\dot{\phi}H \cos a$ ton-ft.

This effect is by far the largest source of torque about the precession axis and is especially large during unstabilized rolling, which tends to complicate the synchronization problem, i.e., the initial stabilizing transient in passing from the unstabilized to stabilizing condition. The torque causes precession in the same direction as the ordered precession and hence tends to overdrive the precession system. It is possible, therefore, that the precession drive could, during some portion of the cycle, become a source of power.

d. Control Design Considerations

The precession velocity of the gyroscopic stabilizer is controlled to provide a moment opposing the roll moment due to waves. Three control signals to be considered are roll angle, roll velocity, and roll acceleration.

(1) A roll angle signal is not possible for control since the stabilizer has zero capacity at zero frequency. This is, it cannot correct for a static list since this would require it to precess indefinitely in one direction.

(2) A roll velocity signal produces a stabilizing moment proportional to roll velocity which would serve in the same sense as the ship's natural damping. Since the ship is inherently a highly underdamped

system, roll velocity is the primary control signal.

(3) A roll acceleration signal could be used in addition to the primary control if improvement of response to high frequency disturbances is found necessary. Since the response of the gyroscopic stabilizer is inherently rapid, such a control would not be of major importance.

Employing the roll velocity signal along, the simplified control equation is:

$$a = \overline{\int \overline{(K\dot{\phi})}\, dt}$$

where:

$k$ = control sensitivity coefficient, (ordered precession velocity/roll velocity)

$k\dot{\phi} = \dot{a}_o$ = ordered precession velocity due to roll velocity, rad/sec

The brackets indicate that limits exist on the precession velocity and precession angle.

e. Stabilizer Capacity

The greatest roll for a periodic waveslope disturbance of a given amplitude occurs when the period of disturbance equals the ship's natural period, $T_s$. For greatest effectiveness under these conditions the gyro precesses in a similar periodic manner, passing through the position of zero precession angle with maximum velocity. For a sinusoidal precession of the gyro at the ship's natural period, the maximum value of the stabilizing waveslope, termed the nominal capacity, would then be:

$$\gamma_s(\text{nominal}) = \frac{\dot{a}_{max} H}{\overline{GM}\ \Delta} = \frac{2\pi}{T_s} \cdot \frac{H}{\overline{GM}\ \Delta} \text{ radians}$$

where:

$a_{max}$ is assumed as one radian

$$\dot{a}_{max} = w_s\ a_{max} = \frac{2\pi}{T_s}$$

$$w_s = \frac{2\pi}{T_s} = \sqrt{K_\phi / J_\phi} = \text{ship's natural frequency of roll, rad/sec}$$

When the $T_s$, GM, and the displacement of a vessel vary with operating condition, the severest requirements on rotor capacity to yield a given nominal stabilizer waveslope occurs when the denominator of the above equation is greatest.

Experience indicates that for disturbances at resonance a stabilizer capacity of approximately 3.5 degrees is generally sufficient. Since the effective waveslope varies inversely as the disturbance period, a nominal capacity (capacity at resonance) of approximately 4.5 degrees would assure adequate capacity for disturbances at periods greater than the ship's natural period. The required rotor angular momentum would be found by solving for H:

$$H = \frac{\gamma_s\ T_s \overline{GM}\ \Delta}{2\pi} \text{ ton-ft-sec}$$

f. Simulation

A functional block diagram of an analog computer simulation of a gyroscopically stabilized vessel is presented in Figure 29.

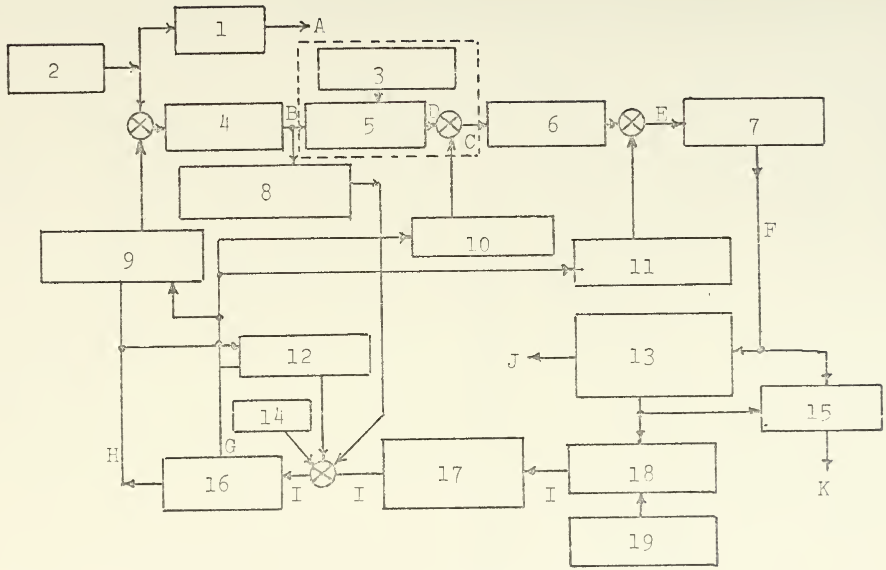



Figure 29. Computer Simulation of Gyroscopically Stabilized Vessel Functional Block Diagram

Figure 29 (Continued)

Conventions

1 Unstabilized Vessel
2 Disturbance
3 Sensitivity Controls
4 Stabilized Vessel
5 Sensing Element
6 Precession Servo
7 Pump Characteristics
8 Ship's Motion Coupling into Precession Axis
9 Stabilizer Characteristics
10 Off-Stroke Characteristics
11 Off Stroke Characteristics
12 Cushions
13 Flow Balance (Leakage Compression Relief Valves)
14 Hard Stop
15 Power Calculation
16 Gyro Dynamics
17 Worm Gear Characteristics
18 Motor Characteristics
19 Synchronization Program

A Unstabilized Ship Motion
B Stabilized Ship Motion
C Control Computer
D Ordered Precession Rate
E Net Stroke
F Pump Flow
G Precession Angle
H Precession Rate
I Torque
J Flow Over Relief Valves
K Power Deliver by Pump

Some features of such a simulation are:

(a) The ability to compare the performance of a stabilized and a unstabilized ship subjected in the same disturbance inputs.

(b) The ability to modify the control sensitivities and other physical parameters, even while the simulation is running, in order to investigate their effect upon stabilizer performance.

(c) The characterization of non-linear relationships; for example, a design feature which mechanically coerces the pump off-stroke before the precession angle limits are reached.

(d) The characterization of the precession hydraulics including fluid compressibility, line pressure, variable relief valve settings, and an indication of valve overflow.

(e) Precession power calculation.

(f) The mechanical efficiency associated with a worm and bull gear precession drive which changes depending upon whether the worm or the bull gear is driving.

(g) The inclusion of all effects upon the gyro: drive inputs, roll-precession coupling, variations of stabilizing moment and coupling effects with precession angle, and the characteristics of hydraulic cushions and hard stops designed to absorb the precession momentum if the precession angle exceeds the normal operating range.

g. Results

There exist some applications of the gyroscopic ship stabilizers. Maybe the most widely known was that aboard the SS Conti di Savoia (1932).

Since then, modern control theory has developed considerably, but very few treatmerts of gyroscopic stabilization have been available. Nevertheless, the results of numerical examples and the better understanding of the ship motion laws, show that the application of gyroscopic roll stabilization to auxiliary ships that operate at low speeds or even at zero speed, like oceanographic vessels, deserves further consideration.

## C. CONCLUSIONS

Roll stabilization as a science has a history of about 90 years. These years have seen the installation of several hundred systems including tanks, gyro-stabilizers and fins. In addition to the sizable amount of engineering effort invested in the foregoing installations a large amount of theoretical and experimental research has been undertaken.

As a result, there now exists a well developed body of theory and data on the subject.

Roll stabilization has been formulated as a control-system problem, closely analogous to problems such as aircraft stabilization, submarine depth control, ship steering etc., and the analysis undertaken here leads to the following specific conclusions:

1. The technical feasibility of stabilization is an established fact. In the future, greater emphasis will be placed on the economic aspects of the problem, reductions of cost volume and weight.

2. The newly developed methods will play an important part in investigations of roll-stabilization since they consider the non-linear aspects of the problem.

3. Several modes of control are possible but at least some feed-back control should always be used. Arguments for feedahead control, predictors, and so on, have largely been negated by advances in control technology.

4. For stabilization at speed of 15 knots, and above, activated fins provide the most efficient stabilization system. Tanks and gyros are worthy of consideration in low speed applications.

5. The principal drawback of both tanks and gyros is that they are not suitable for stabilization against very low frequencies and yaw-heel effects. However inputs of this type are not present in low speed applications.

6. In stabilization by activated fins, it is necessary to consider false angle-of-attack inputs at the fins as well as the basic effective waveslope and drift-angle inputs. These false angles have an important effect on heavy weather performance.

7. These effects may be eliminated by repeating back fin lift instead of fin angle against the order from the control amplifier. This also provides a means for limiting

the maximum load on the fin, and decreases the expenditure of stabilizing power.

8. The computer is a powerful tool in analysis of roll stabilization and similar systems. It is particularly useful in studying economic questions, which almost always involve essential nonlinearities.

## V. PROPULSION SYSTEM

### A. PROBLEM

As propulsion plants and control systems are improved and put into operation, an increasing degree of automation is being sought and utilized. For naval ship-propulsion systems, a great many different propulsion plants and combinations of these plants are under investigation. The reason for the large variety of plants being evaluated is the broad range of propulsion plants needed in naval applications. Each of these can perform its particular mission effectively, only if the size, weight, speed, endurance, and maneuverability of the vessel are optimized.

The propulsion plant of a modern naval ship is a complex assortment of systems which form the parts of the steam and propulsion power cycles. These systems and their subsystems generally exhibit some degree of interaction, and the total performance of the plant is dependent upon proper functioning of each system and each element in each system. It seems evident, therefore, that the automation of all or any part of the steam cycle elements must be approached from the steam cycle elements must be approached from the system's engineering point of view. This is in fact believed to be the cardinal rule for the starting point of any design effort relating to the control of any process containing interacting systems whose complexities are such that they cannot be classified as trivial. The control designer,

when working in a propulsion plant, can not afford to think only in terms of the design requirements of his specific item of interest (such as a boiler, pump, desearator or forced draft blower). On the countrary, he must consider the dynamic compatibility with the other elements of the system, and he must be willing to modify an otherwise good design in order to achieve optimum total system performance.

The most important consideration in the overall system design is a recognition of the nature of process dynamics. A knowledge of the characteristics of the process is essential to the production of a satisfactory control system. These nonlinearities result from the inherent working principles of the machinery, the physical nature of the fluids in the process, and the limitations of the control devices themselves. In any case, the effect of these non-linearities on the performance of the closed-loop system should be considered.

Once a control system design has been proposed, it should be completely tested for its stability and transient response characteristics. These tests may be accomplished either by classical linear systems analysis techniques or by the use of an analog or digital computer simulator. Experience with complex and sophisticated control systems indicates that these efforts are generally economically justified.

The first part of the following section will cover the description of some of the most important propulsion systems

using automatic control that we can find today on board Naval ships. The second part will be devoted to the review of certain recurrent problems in the control of naval machinery.

## B. ANALYSIS

The propulsion system chosen for any naval ship must be a logical answer to its design capabilities and to the requirements and characteristics that its mission and purpose will imply. As consequence we can observe a big proliferation of systems and combination of systems pretending to find an optimum performance.

Some of those systems which may represent the more important applications actually in use to the automatic propulsion plant, will be described. Detailed information and transfer functions can be obtained from classified references.

To facilitate the analysis and to give some definite points of comparison the automated propulsion plants discussed in this section are intended to be generally suited for, but not restricted to use in a single screw combatant type naval vessel.

## C. COGAG ELECTRIC PROPULSION PLANT

### 1. General

Typical naval operations such as: antisubmarine warfare, minesweeping, convoy escort, etc., require ships

to operate at speeds considerably lower than maximum design speed. The combined power plant seems ideally suited to propulsion of a ship of those characteristics. The power to drive a surface ship varies roughly as the cube of its speed. Therefore at 1/2 to 2/3 speed only 1/8 to 8/27 of the power is needed. High speed naval ships of 30 to 40 knots capability operate in their 1/2 to 2/3 speed range the major portion of their time. They operate at greater than 1/4 to 1/3 power only 15 - 20% of their under way time. The combined plant concept to meet this kind of operation uses a high-efficiency, long-life base plant for the frequently used cruising speeds. It is then supplemented by lightweight, perhaps less efficient, perhaps short time between overhauls type of boost plant.

The obvious choice for the lightweight, high power boost plant is the marinized, aircraft-type gas turbine. This engine essentially amounts to an aircraft jet engine, modified somewhat for extended service in a marine environment, and coupled to a free power turbine which converts the gas flow to shaft rotation. The principal advantages are extremely high power density and fast response.

The choice of a base plant is not so straightforward with diesel, steam, and various forms of gas turbines all having certain advantages and disadvantages.

Gas turbines are unidirectional, high speed machines while a propeller operates at low speed and must produce

bidirectional thrust. There are a wide variety of techniques for accomplishing the transition.

Speed reduction may be accomplished by reduction gears or electric drive, but the choice is intimately connected with the reversing problem. Possible reversing methods include reversing gas turbines, reversing gears, controllable-reversible pitch (CRP) propeller, and electric drive. The reversing gas turbine is still in the feasibility study stage. Electric drive is available and well proven. Unfortunately its components are large and heavy. The space and weight requirements become prohibitive if both base and boost plants employ electric drive.

Any solution represents some compromise. It appears that the best compromise at the present time is the scheme shown in Figure 30 which employs reversible electric drive for the base plant and nonreversible gearing for the boost plant. This represents a transmission system which, while far from optimum, is at least realizable with minimum risk within the state-of-the-art. A minor variation of this scheme would employ a high speed motor driving through the reduction gear rather than a low speed motor driving the shaft directly. While this would permit some space and weight saving, it would require that the reduction gear, which is a prime source of sonar interfering noise, be in the drive train at all speeds.

The gas turbine is relatively easy to automate, being largely self-contained and with only moderate

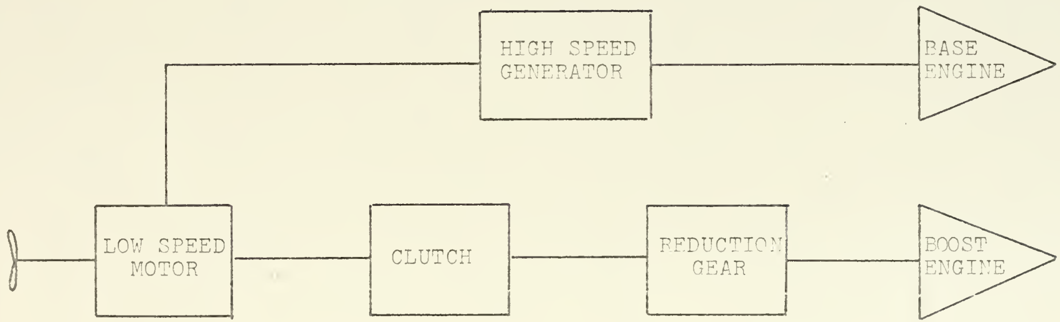



Figure 30. COGAG Electric Propulsion Plant. Power Transmission

requirements for external auxiliary systems. Having received most of its developments both from the aircraft industry and for industrial applications where long term unattended and automatic operation was necessary, it necessarily evolved into a highly reliable and automated power plant. Another factor contributing to the advanced state of automation of gas turbines is the nature of the gas turbine itself which, with its very fast response, high performance capability and dependence on environmental conditions makes automatic control mandatory for safety and consistently high performance.

Solid state electronics for sensing, logic and command functions with electrohydraulic and servomotor control components has been chosen for the control system. Both base and boost plant gas turbine control systems are electronic at the sensing-logic/compute-command level. Electrical command signals cause actuation of electro-hydraulic or electromechanical elements.

To provide for immediate local control of both the base and boost gas turbines and provide an increased measure of reliability and independence from general system failure, separate engine control systems independent of a central propulsion plant signal processor are required.

The following Cogag Installations will be described:

a. Cogag Electric Propulsion Plant using a regenerative base turbine

b. Cogag Electric Propulsion Plant with simple cycle turbines

c. Cogag Propulsion Plant with reversing gear.

2. <u>Cogag Electric Propulsion Plant with Regenerative Base Turbine</u>

The base plant is a regenerative, gas-turbine (about 16,000 hp) electric plant which consists of an a-c propulsion generator connected to a low-speed multipole motor directly driving the shaft and fixed pitch propeller. The boost plant consists of a simple single-cycle aircraft type gas turbine rated at about 25,000 hp, which is clutched to the shaft through a reduction gear. The total shaft power is about 38,000 horsepower.

Direct throttle control over both turbines from the bridge, subject to control transfer and override by the engine room, is utilized. Other automated subsystems include automatic start-shutdown programmers for both gas turbines; fully automatic electric propulsion-plant operation in starting, stopping, and reversing; fully automatic gas-turbine and combined plant throttle-control systems; and automated engine auxiliary systems.

A propulsion-plant control system block diagram appears as Figure 31. This diagram shows the interrelationships between the major propulsion-plant machinery systems and the control signals. The principal systems operated and monitored by the EOS (Engineering Operations Station) are also indicated.

The design approach is one of centralized propulsion plant control from the Engineering Operations Station

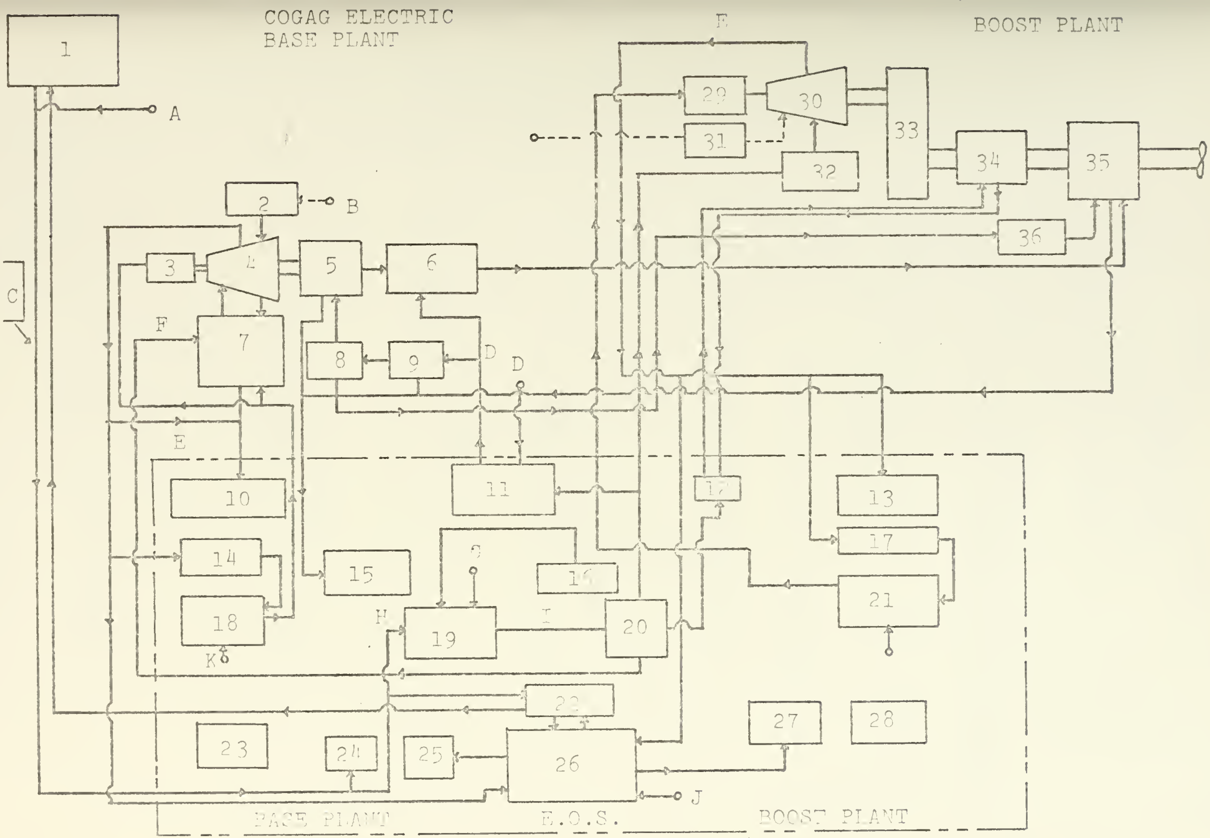



Figure 31. Automated Cogag Propulsion Plant

Figure 31 (Continued)

Conventions

1 Bridge Propulsion Control Console
2 Hydro Mechanical Control System
3 Turbine Starter
4 Regenerative Gas Turbine
5 Base Plant Propulsion Generator
6 Forward and Reverse Disconnect Contactors
7 Electronic/Electrohydraulic Regenerative Gas Turbine Control System
8 Field Exitation -Generator
9 Field Exitation Regulator
10 Regenerative Gas Turbine Instrumentation Including L.O., F.O., and Air Systems
11 Start-Stop-Reverse Sequencer
12 Clutch Control
13 Boost Gas Turbine Instrumentation Including L.O., F.O., and Air Systems
14 Vibration Analyzer
15 Electric Propulsion Plant Instrumentation
16 Control Transfer Switches and Displays
17 Vibration Analyzer
18 Regenerative Turbine Start/Shutdown Programmer
19 Control Transfer Logic
20 Combined Plant Throttle Control System
21 Boost Gas Turbine Start and Shutdown Programmer
22 Plant Set-Up Control and Displays
23 Base Plant Auxiliary Machinery Controls and Instrumentation
24 Auto Bell Logger
25 Auto Prop. Plant Data Logger
26 Propulsion Plant Signal Processor
27 Alarms and Displays
28 Boost Plant Auxiliary Machinery Controls and Instrumentation
29 Turbine Starter
30 Boost Gas Turbine
31 Hydromechanical Control Systems
32 Electronic Electrohydraulic Control System
33 Boost Plant Reduction Gear
34 Clutch
35 Low Speed Synchronous Propulsion Motor
36 Field Exitation Control
A From C.I.C. or Auxiliary Station
B Emergency Local Throttle Control
C Power Plant Control Signals
D Shaft, Clutch and Turbine Status Signal
E Instrumentation Signals
F Throttle Control Signals
G E.O.S. Throttle
H Bridge Throttle
I Throttle Signal
J Misc. Propulsion Plant Status Signals
K Interlock and Status Signals

and complete automation of gas-turbine control systems and auxiliaries to the maximum practical extent.

To provide for immediate local control of both the base and boost gas turbines and provide an increased measure of reliability and independence from general system failure, separate engine control systems independent of a central propulsion-plant signal processor are required.

While integration of all the logic operations, interlocks, and automatic control signals issued to actuators and engines could be accomplished by a propulsion-plant central computer, this has not been attempted because of the emergency need for independence and human judgment at the decision-making level.

It should be recognized that this approach still involves considerable subsystem automation beyond the Engineering Operations Station. To a large extent, the plant is then "automated." It is mandatory for a combatant ship that local controls be available in all subsystems in case of failure or damage to either the Engineering Operations Station, automatic control systems, or actuators.

The ship's shaft rpm is the basic controlled parameter for the propulsion plant. Shaft rpm is ordered either at the bridge or the Engineering Operations Station, and the propulsion control system issues turbine commands in a closed-loop feedback control system to achieve the desired rpm.

The Bridge is given control by the Engineering Operations Station, which always exercises an override capability on the Bridge.

Control of both gas turbines is accomplished by the combined plant throttle control system at the Engineering Operations Station which issues electrical signals to the electronic/electrohydraulic control systems of both gas turbines. The propulsion-plant throttle control sets the required shaft rpm from either the Engineering Operations Station or the Bridge, and the combined plant throttle-control system issues individual throttle commands for each gas turbine as required.

Emergency local throttle controls are backup equipment for the gas turbines. These mechanical controls consist of governors and fuel control computer elements. The additional complexity added to the system by the backup mechanical control system is very slight.

Shaft speed orders which involve starting, stopping, or reversing the direction of the shaft cannot be executed by the gas-turbine throttles alone, since manipulation of the electric drive train is also required. Normally, these operations are carried out automatically by the "start-stop-reverse sequencer" which controls the electric plant and the turbine throttles.

A brief description of the subsystems corresponding to a typical installation follows. (Block Diagrams are found in the publications of Naval Ship Research and

Development Center, Marine Engineering Laboratory, listed in the references.)

a. The Regenerative Gas Turbine

The Regenerative Gas Turbine is a single spool-compressor type with inlet guide vane positioning (to control compression ratio) and turbine nozzle area control to maintain a relatively constant turbine inlet temperature and improve part load performance. Recuperator bypass and compressor bleed valves are also incorporated. The primary Regenerative Gas Turbine electronic/electrohydraulic control system consists of the following four subsystems: power turbine speed governing, turbine inlet temperature control, overspeed control, and compressor control.

(1) R.G.T. Instrumentation. The RGT instrumentation provided is typical of fixed-base and marine gas-turbine installations. The instrumentation includes signals provided for status monitoring at the EOS ($N_{GS}$) and signals provided to automatic control systems ($N_{GC}$).

(2) R.G.T. Hydraulic Starting System. The RGT starting system utilizes two variable-delivery hydraulic pumps driven by two 100-hp induction motors powered by the ships' service electrical system. The high-pressure oil is fed to a variable-displacement hydraulic starter mounted directly on the gas turbine. Either pump or both can supply adequate hydraulic pressure for starting the gas turbine. In normal operation, it is expected that one of the hydraulic pumps will be used to start the base turbine

and the other to start the boost turbine, thus sharing a common hydraulic system. Turbine cranking is automatic with solenoid activation of the hydraulic pump and automatic cutout when the turbine is ignited and the proper speed is reached.

(3) R.G.T. Automatic Start/Shutdown Programmer. The automatic start/shutdown programmer accepts a start signal from either the EOS, turbine room, or bridge (for the boost turbine) and begins a programmed starting sequence subject to interlocks, together with turbine and auxiliary system status signals through a programmed sequence of starting operations. Failure of the turbine to respond to a particular operation within a predetermined interval or set of conditions interrupts the sequence and an alarm is given. After successful start, the turbine throttle is transferred to the idling control, and the turbine receives an idling fuel rate signal until a signal is sent to the idling control to transfer from idling to an ordered throttle signal. Normal shutdown is accomplished by allowing 5 minutes at an idling speed before the fuel valve is closed to shut down the turbine.

(4) R.G.T. Hydromechanical Control System. As a backup system for both the RGT and boost gas turbines, a separate hydromechanical control system is used. Manual starting and operation of the gas turbine in the backup mode are from a secondary turbine control panel located in the turbine room. This backup subsystem includes hydromechanical

fuel control and throttle, very essential RGT instrumentation, manual start controls, and other manually operated RGT controls.

(5) <u>R.G.T. Lubricating and Fuel Oil Systems</u>. The RGT lubricating oil system is a self-contained subsystem which is integral with the gas turbine except for oil coolers and a sump or storage tank. The fuel oil system is extremely simple. Control and monitoring of both oil systems for the RGT are accomplished from the EOS.

b. The Electric Propulsion System (Figure 32)

The electric propulsion subsystem consists of an alternating-current generator, exciters, alternating current synchronous motor, and the necessary switch gear. The subsystem couples the base engine to the shaft while permitting a reversing of the unidirectional prime mover output.

The conventional three-phase, alternating current generator is connected directly to the output shaft of the base plant power turbine. It is a totally enclosed, air-cooled unit with integral saltwater air cooler. This unit is capable of absorbing the full 16,000 bhp output of the base engine.

Generator and motor excitation is provided by motor-generator sets operated from ships' service alternating current power. Two of these units are provided, with one in operation and one for standby use. Under normal operating conditions, one unit supplies both motor and

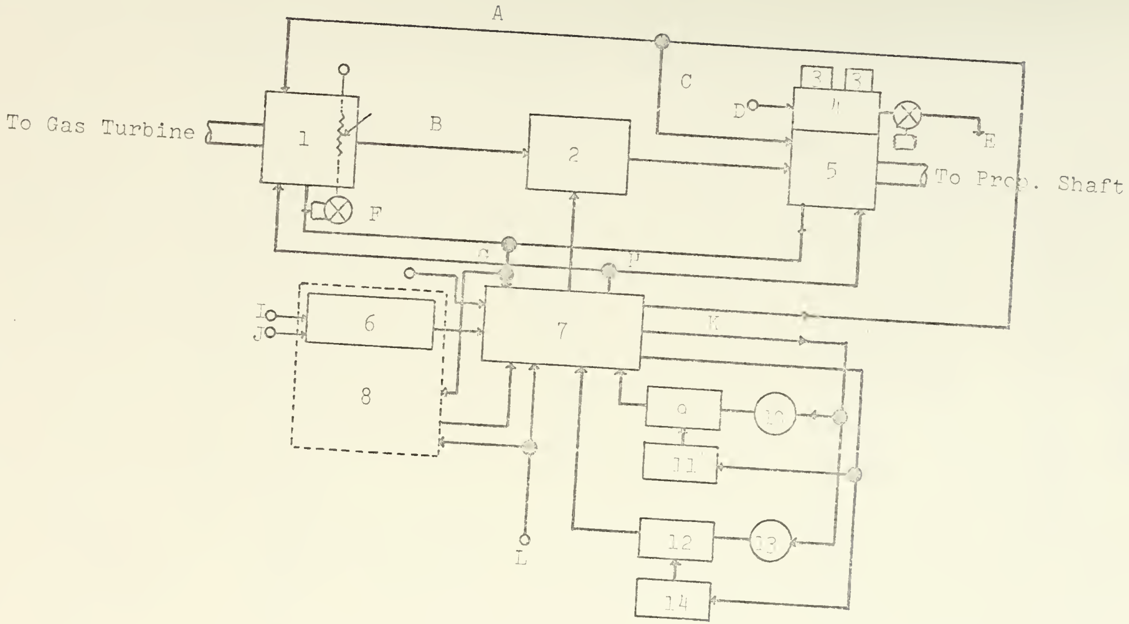



Figure 32. Cogag Electric Propulsion Plant. The Electric Propulsion System

Figure 32 (Continued)

Conventions

1 Propulsion Generator
2 Disconnect Forward Reverse Contractor
3 Cooling Fans
4 Heat Exchanger
5 Propulsion Synchronous Motor
6 Start/Stop Reverse Sequence
7 Electric Propulsion Control Panel
8 Electric Propulsion Plant Display, Alarms & Records
9 Exitation Generator
10 Motion
11 Exitation Regulator
12 Exitation Generator
13 Motor
14 Exitation Regulator

A Anti-Condensation Heater Signals
B Integral Heat Exchanger
C Water/Air Cooler
D Sea Water Level
E S. W. Discharge
F S. W. Discharge
G Instrumentation Signals
H Generation Motor Field Exitation
I Shaft, Clutch, Turbine & Status Signals
J From Combined Plant Throttle Control System
K Motor Power Exiter Control
L Instrumentation Signals from Exiters

generator excitation power. It is also possible to provide separate excitation for the two units.

The propulsion motor is a three-phase, alternating current, induction-start, synchronous-run type rated at 14,500 bhp output. Like the main generator, this is a totally enclosed unit with an integral heat exchanger. Since this motor is directly coupled to the shaft, it is a low-speed multipole motor, and the speed reduction from the base turbine to shaft speed is accomplished in the pole ratio between the propulsion generator and the propulsion motor. Motor excitation is provided by the motor-generator sets as discussed above.

The connecting switch gear provides high power level switching facilities for motor reversing which is accomplished automatically by the start-stop-reverse sequencer.

(1) Propulsion Generator and Motor. Separate motor-driven pumps are available to supply lubricating oil to the generator without operation of either the propulsion motor or the reduction gear. In addition, separate oil temperature and pressure monitoring and alarms are used for the propulsion generator-motor system. From Figure 32 it may be noted that electric propulsion plant control and instrumentation is possible from both the electric propulsion control panel (EPCP) and the Engineering Operations Station. The EPCP is a normally unmanned station used for

detailed analysis and local control action of the electric propulsion system.

(2) Electric Propulsion Plant Start-Stop-Reverse Sequencer. Normal engine orders are executed by the base and boost turbine speed controls. However, engine orders which involve changing the direction of shaft rotation cannot be accomplished by turbine speed variations. Such orders include ahead-to-astern (including crash reverse), astern-to-ahead, stop-to-either-direction, and either-direction-to-stop, all of which require control of the base turbine and electric propulsion plant. The start-stop-reverse sequencer is primarily digital in nature and provides all the automatic control of the electric propulsion plant, which includes interlocks on the boost plant clutch and throttles as well as motor-generator field excitation, main contactor closure, and reversing and control of the base plant throttle signal.

c. Boost Gas Turbine

The primary control of the Boost Gas Turbine is an electronic/electrohydraulic control system. Its instrumentation, the Hydraulic Starting System and the automatic Start/Shutdown Programer are essentially the same as for the regenerative gas turbine except that the boost gas turbine is single cycle, employs a twin spool compressor, and does not have either inlet guide vane positioning or turbine nozzle area control. In addition, compressor bleed air is not used.

d. Common Propulsion Auxiliary Systems

(1) Turbine Water-Wash System. The need to water wash gas turbines periodically is well established as a means of removing deposits accumulated in a saltwater environment. In the interest of crew reduction and simplification of the procedure required, a permanently installed semiautomatic water-wash system is required.

Distilled water from the ship distilling plant is stored in a tank. A motor-driven pump distributes the water to the base, boost and ship service generator turbines. The turbine to be cleaned is cranked by means of the starting system, and water is sprayed into the compressor inlet from installed spray nozzles.

(2) Fuel Oil Storage, Transfer and Service System. The fuel Oil Storage System is a continuous ballasting compensated system. It permits the ship to maintain maximum stability at all times without close attention from operating personnel. The compensated fuel-oil ballast tanks in each group are piped in a cascade arrangement with sluice valves between each adjoining tank. When suction is taken on a tank amidships, the fuel is replaced by fuel from the adjoining tank. The total system volume is maintained by introducing salt water into the last tank of the series.

The transfer system consists primarily of two motor-driven transfer pumps and two centrifuges. The

pumps provide the means for transferring fuel oil from the stowage tanks to the service tanks.

The service system comprises the two service tanks and the equipment and piping necessary to provide a continuous supply of clean fuel at a positive pressure to the propulsion turbines and the ship service generating system.

The service system can be operated as one loop or segregated into two separate loops as a damage control measure. Three motor-driven, positive-displacement pumps are provided. Two are of adequate capacity to supply the entire plant demand. In single-loop operation, one is in service with the other as standby. The third pump is a low-capacity unit for light loads and for port use to avoid excessive recirculation. Each pump discharges through a fine mesh basket-type duplex strainer and coalescing-type filter/separator to the fuel-oil headers.

Under normal conditions, the fuel-oil systems are intended to be operated remotely from the Engineering Operations Station; and remote controls for all pumps, valves, and other equipment which are involved in normal operation are provided there, as well as the required instrumentation to ensure proper operation.

(3) Reduction Gear and Associated Main. Lubricating Oil System. The main lubricating oil system provides lubrication to the reduction gear, propulsion generator and propulsion motor. This system is quite

similar to that used in a conventional steam-turbine plant. Three positive-displacement pumps, taking suction from the sump through coarse strainers, provide pressurized oil under all operating and emergency conditions.

Under normal circumstances, the oil system is self-regulating and operates automatically without operator attention. Instrumentation and control facilities are provided at the Engineering Operations Station to permit system start-up and securing, to monitor system operation, and to take limited casualty control action. Local regulation is provided, where possible, for design simplicity.

e. Combined-Plant Throttle-Control System (Figure 33)

Both the Engineering Operations Station and the Bridge throttle controls set the ordered propeller shaft rpm into the combined-plant throttle-control system.

Starting of the base plant (RGT) is initiated from the Engineering Operations Station which begins the starting cycle of the automatic starting/shutdown programmer. After starting is successfully completed, the RGT receives its throttle signal from the idling control, a portion of the automatic starting/shutdown programmer. The RGT remains idling until the idling control receives a signal from the Engineering Operation Station to transfer the RGT throttle to the throttle control. At this point, either the Engineering Operation Station or the bridge

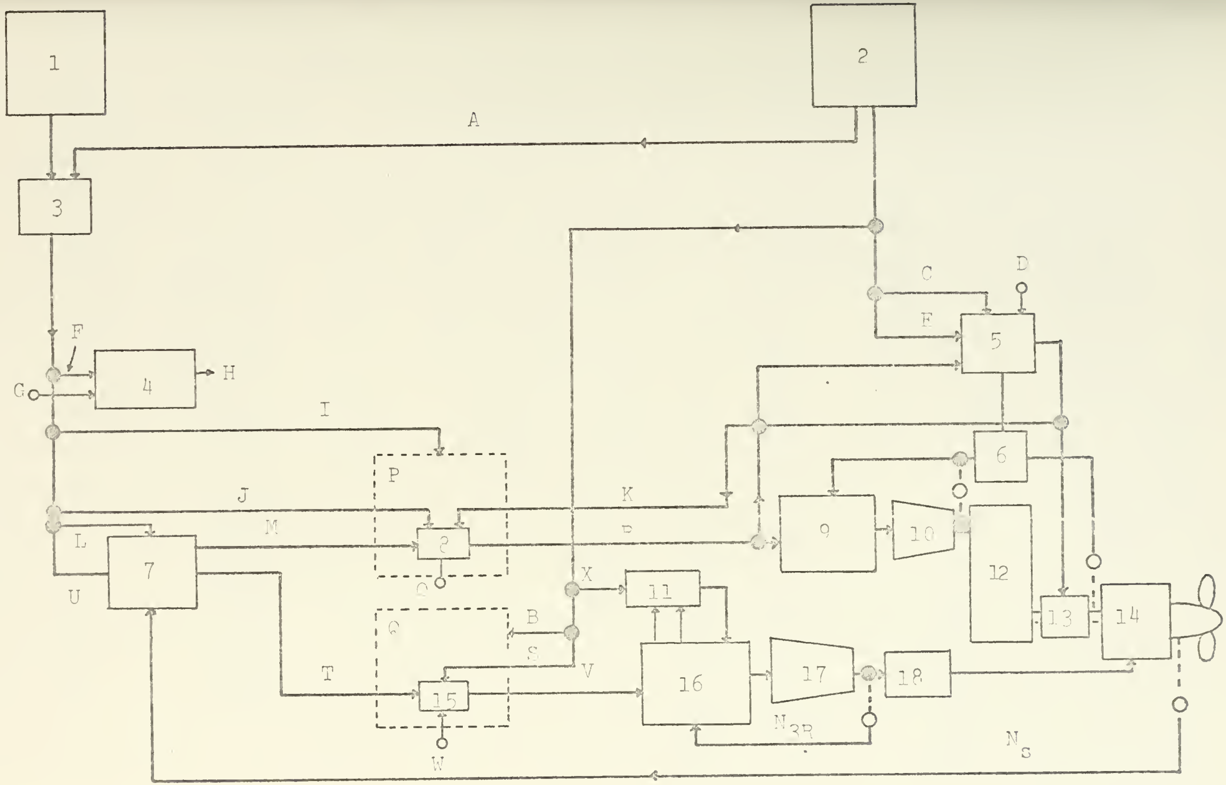



Figure 33. Cogag Electric Propulsion Plant. Combined Plant Throttle Control System

Figure 33 (Continued)

Conventions

1 Bridge Propulsion Control Console
2 E.O.S. Propulsion Control Console
3 Control Transfer Logic
4 Start, Stop, Reverse Sequencer
5 Clutch Control
6 Differential R.P.M. Sensor
7 Shaft Error Signal Conditioner
8 Idling Control
9 Electronic/Electrohydraulic B.G.T. Control System
10 Boost Gas Turbine
11 Max. Power Detector
12 Boost Plant Reduction Gear
13 Clutch
14 Propulsion Motor
15 Idling Control
16 Electronic/Electrohydraulic R.G.T. Control System
17 Base Regenerative Gas Turbine
18 Propulsion Generator

A Engine Control, Control Transfer and E.O.S. Override Signals
B Base Engine Start and Shutdown
C E.O.S. Interlock and Control
D Clutch Disengage from S.S.R. Sequencer
E "Clutch in Speed" and Hysteresis Adjust
F Shaft Direction Signal
G Status Signal from Electric Plant
H Electric Plant Control Signal
I Boost Engine Start/Shutdown
J Boost Engine Throttle Transfer
K B.G.T. Idling Signal
L Sea State Adjust
M Ordered $N_{3B}$
$N_{3R}$ Power Turbine R.P.M. (R.G.T.)
$N_{3B}$ Power Turbine R.P.M. (B.G.T.)
$N_s$ Shaft R.P.M.
O Override Idling Signal from S.S.R. Sequencer
P Automatic Starting/Shutdown Programmer B.G.T.
Q Automatic Starting/Shutdown Programmer R.G.T.
R B.G.T. Throttle Signal
S Base Engine Throttle Transfer
T Ordered $N_{3R}$
U Ordered Shaft R.P.M. ($N_s$)
V R.R.T. Throttle Signal
W Override Idling Signal from S.S.R. Sequencer
x Max. Power Set

throttle becomes "active" and the RGT receives its throttle signal via the bridge shaft-speed or Engineering Operation Station shaft-speed command. The ordered shaft rpm ($N_s$) is an input to the shaft error-signal conditioner which compares ordered and sensed shaft speed and issues the throttle signals to both engines. The error signal conditioner contains all of the circuits necessary to match the desired linear $N_s$ command to issued throttle signals, provide for adjustment in loop response via the sea-state adjust, and provide for open-loop shaft control when desired (such as in high sea-state conditions). Shaft direction is interpreted and electric plant reversing accomplished by the start-stop-reverse sequencer.

Operation to the maximum cruising speed is possible with the RGT and the electric propulsion plant. For speeds in excess of the maximum base plant cruising speed, additional power beyondthe 16,000-hp limit of the RGT is necessary and is provided by the boost turbine. When the maximum power limit of the RGT is exceeded, the RGT maximum power detector issues a signal to the RGT control system to interrupt the constant speed governing control and impose a fixed power limit on the RGT. Thus, the RGT is no longer responsive to shaft-speed commands and instead provides a fixed-power output by means of fuel-flow rate limiting. An indication is given at the Engineering Operation Station and bridge that the boost turbine, if not idling, must be started. The boost turbine

is started by its automatic starting/shutdown programmer with a signal from the Engineering Operation Station setup control or from the bridge setup control subject to an Engineering Operation Station override through its own interlock. After a successful start, the boost-turbine throttle is transferred to the idling control. Transfer to ordered boost-turbine rpm ($N_{3B}$) is accomplished at the Engineering Operation Station or at the Bridge subject to a transfer approved from the Engineering Operation Station. The boost-turbine throttle signal is provided by the shaft-error signal conditioner in response to ordered shaft speed. Boost-turbine assistance provides the additional power required to satisfy the required rpm in the speed range from cruising to full speed.

Boost-turbine clutching is accomplished automatically through the clutch control. A friction clutch is assumed; this requires an "engage" signal. A differential sensor compares the speed on both sides of the clutch and provides a signal "OK to engage clutch." In addition, an Engineering Operations Station "OK to engage clutch" is necessary for clutch engagement which occurs automatically as the boost-turbine speed reaches the "clutch-in speed" set into the clutch control at the Engineering Operation Station.

3. Cogag Electric Propulsion Plant with Single-Cycle Turbines

This plant is basically the same as the Cogag Electric Propulsion Plant with Regenerative Base Turbine

except that a single cycle turbine is substituted for the regenerative base turbine. Assuming a horse-power rating of 21,500 for the single-cycle turbine in the base application and assuming typical electric plant efficiencies, the total combined plant power is about 44,000 shaft horse power (shp).

The base turbine drives an a-c propulsion generator, which dirves a low-speed, synchronous 20,000-hp motor directly coupled to the shaft. Several alternate arrangements in the propulsion motor might be necessary because of the size and weight of a 20,000-hp, low-speed motor. For example, a higher speed synchronous motor of the same horsepower (or two 10,000-hp motors) could be operated through the reduction gear.

4. Cogag Propulsion Plant with Reversing Gear (Figure 34)

Two single-cycle gas turbines are used in this combined plant, which (except for the electric propulsion system) is identical in all respects to the two previously discussed plants. Assuming a power rating of 21,500 shp for the gas turbine used in the base plant application and 25,000 shp for the gas turbine in the boost mode, and assuming typical reduction gear efficiencies, the total power available to the shaft is about 45,000 shp.

Reversing is accomplished with a reversing/reduction gear with friction clutches for forward and revers operation. The reverse power is provided by the base turbine only.

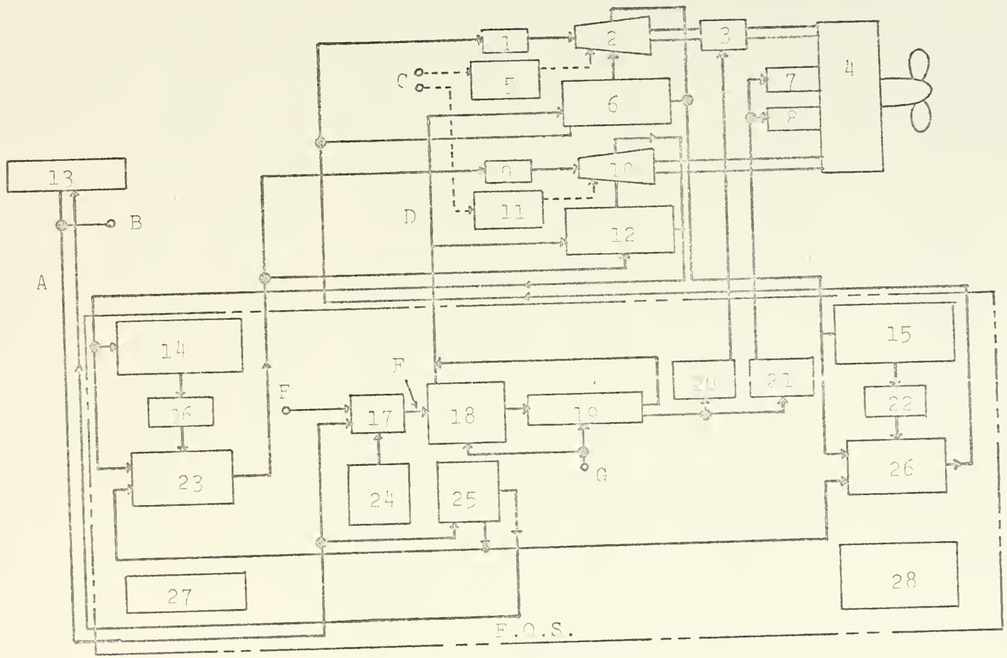



Figure 34. Automated Cogag (with reversing gear) Propulsion Plant

Figure 34 (Continued)

Conventions

1 Turbine Starter
2 Boost Gas Turbine
3 Boost Clutch
4 Main Reversing Reduction Gear
5 Hydromechanical Control System
6 Electronic Electrohydraulic Control System
7 Forward Clutch
8 Reverse Clutch
9 Turbine Starter
10 Base Gas Turbine
11 Hydromechanical Control System
12 Electronic Electrohydraulic Control System
13 Bridge Propulsion Control Console
14 Base Gas Turbine Instrumentation, Including LO.FO and Air Systems
15 Boost Gas Turbine Instrumentation, Including LO.FO and Air Systems
16 Vibration Analyzer
17 Control Transfer Logic
18 Combined Plant Throttle Control Systems
19 Start, Stop, Reverse Sequencer
20 Boost Clutch Control
21 Forward Reverse Clutch Control
22 Vibration Analyzer
23 Base Turbine Start Shutdown Programmer
24 Control Transfer Switches and Displays
25 Plant Set-Up Controls and Displays
26 Boost Turbine Start Shutdown Programmer
27 Base Plant Auxiliary Machinery and Instrumentation
28 Boost Plant Auxiliary Machinery Controls and Instrumentation

A Plant Control Signals
B From CIC. or Auxiliary Station
C Emergency Throttle Controls (local)
D Throttle Control Signals
E EOS. Throttle
F Throttle Signal
G Shaft, Clutch and Turbine Status Signals

The reversing/reduction gear uses an idler gear to provide reverse rotation and two friction, pneumatically actuated clutches which are engaged for either forward or reverse operation of the shaft. These clutches are air cooled and can absorb the large amount of energy involved in clutching from a high-ahead speed to a crash stop.

The boost turbine is connected to the reduction gear through a separate boost-turbine clutch and is not reversible, whereas the base turbine is disengaged from the gear through the reversing/reduction gear clutches.

The main difference between this system and the above Cogag Systems are the additional complexity of the reversing/reduction gear and the reduced complexity in the start-stop reverse sequences.

In addition, the electric propulsion system is absent in this plant, a factor contributing to a large decrease in complexity. All the other subsystems are either the same as the first two plants studied or do not change significantly.

## D. CODAG PROPULSION PLANT (Figure 35)

The Codag propulsion plant consists of a 12,000-hp diesel engine base plant and a single gas-turbine boost plant. Assuming a rating of 25,000 hp for the gas-turbine in a boost application, and using typical reduction gear efficiencies, the total combined plant power is about 36,000 shp.

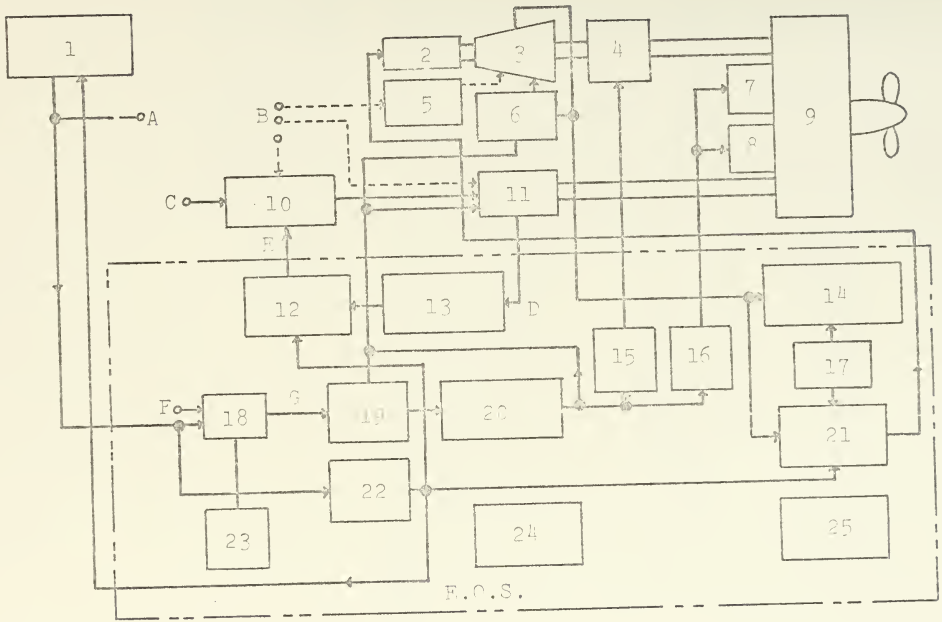



Figure 35. Automated Codag Propulsion Plant

Figure 35 (Continued)

Conventions

1 Bridge Propulsion Control Console
2 Turbine Starter
3 Boost Gas Turbine
4 Boost Clutch
5 Hydro-Mechanical Control System
6 Electronic Electro-Hydraulic Control System
7 Base High Speed Clutch
8 Base Low Speed Clutch
9 Main Reduction Gear
10 Air Start and Braking Controllers
11 Diesel Engine
12 Diesel Engine Start Shutdown Programmer
13 Base (diesel) Engine Instrumentation
14 Boost Gas Turbine Instrumentation
15 Boost Clutch Control
16 Base Clutch Control
17 Vibration Analyzer
18 Control Transfer Logic
19 Combined Plant Throttle Control System
20 Start-Stop-Reverse Sequencer
21 Boost Turbine Start/Shutdown Programmer
22 Plant Set of Controls and Displays
23 Control Transfer Switches and Displays
24 Base Plant Aux. Machinery Controls and Displays
25 Boost Plant Aux. Machinery Controls and Displays

A From C.I.C. or Auxiliary Stations
B Emergency Local Controls
C From Starting Air Supply
D Instrumentation Signals
E Throttle Control Signals
F E.O.S. Throttle
G Throttle Signal

For a destroyer escort, the estimated speeds with this power plant would be about 20 knots maximum on the diesel engine and a top speed of about 28 knots with the diesel and gas turbine combined.

The diesel engine is designed as a medium speed engine that requires a two-speed reduction gear, a low speed to be used only when the diesel is running alone and a high speed gear when the base plant is running combined with the gas turbine.

Reversing is accomplished by shutting down the diesel engine and starting it in the reverse direction. For crash reverse, dynamic air braking of the diesel engine is used to bring the shaft to a rapid stop, and then the diesel engine is started in reverse. The base plant clutches can be used to disconnect the diesel engine from the (pass) reduction gear when necessary in reversing operations. An additional control mode useful in entering and leaving port is the split-plant mode in which the diesel engine is running in the reverse direction and the boost turbine is idling. Forward and reverse can then be executed very rapidly by alternately clutching and declutching the diesel and gas-turbine clutches.

1. Diesel Engine Control System

The primary diesel-engine control element is a modified conventional electro hydraulic governor. In the base speed range (closed-loop shaft-speed control of the diesel), the governor accepts an ordered speed signal in

electrical form from the combined plant throttle-control system and a signal representing actual engine speed to generate the proper fuel injector rack position to maintain the ordered speed. At speeds above the base range where shaft speed is controlled by the boost engine, the governor operates the base engine according to a preprogramed relationship of fuel rack position versus speed. Manual backup engine control is by means of a centrifugal governor which is integral with the normal electrical governor and so arranged that failure of the electrical system automatically shifts control to the backup. The mechanical governor normally "tracks" the electrical governor at a slightly higher speed, using the same input signal. A separate over-speed system is provided which shuts down the engine fuel supply in the event failure of both governors.

2. Diesel Engine Starting Air System

This system consists of two 600 psi multistage air compressors with saltwater cooled intercoolers and aftercoolers supplying a series of air receivers which have a capacity to permit several normal engine starts. The compressors are operated so as to start automatically in sequence on receiver pressure drop.

The diesel engine is started either at the engine of remotely from the Engineering Operations Station by means of the diesel-engine automatic start/shutdown programmer

which controls the diesel cam position and air start valves.

3. Diesel-Engine Automatic Start/Shutdown Programmer

This unit is functionally identical to that used with the gas turbine, but it has the following significant differences in operation:

1. Initiation of the starting sequence automatically from the start-stop-reverse sequencer as well as manually from remote locations.

2. The requirement for selecting the direction of rotation while starting.

3. The substitution of direct air starting.

4. The deletion of the ignition system and a number of timed sequences relating to it.

5. The absence of the requirement for a short idling period before shutdown.

4. Diesel Engine Lubricating Oil System

Lubricating oil is supplied to the engine supply header by an engine-driven pump which takes its suction from the sump at the base of the engine. The oil goes through a reversing-valve chest to provide a unidirectional oil flow for both directions of engine rotation. The pump discharges into a duplex strainer which discharges into the cooler where the oil is cooled by jacket water before being supplied to the engine header. The oil temperature is automatically maintained by regulation of the water jacket temperature.

A motor-driven prelubrication pump is run in parallel with the engine-driven pump, bypassing the reversing-valve chest. This pump supplies oil for prelubrication of the engine after an extended shutdown.

Oil which has been contaminated by combustion products is collected by a scraper-oil ring and drained to the scraper-oil drain tank. Crankcase oil is mixed with the cylinder oil in the oil drain tank and discharged by a motor-driven pump through a scraper-oil filter back into the engine crankcase. This is a continuous bypass oil purification system.

The cylinders are lubricated by engine-driven mechanical lubricators supplied by an engine-driven cylinder oil supply pump. This pump includes a built-in reversing mechanism to supply undirectional oil for both engine directions and takes suction from the diesel-engine cylinder lubricating oil service tank.

5. Diesel Engine Freshwater and Saltwater Cooling Systems

Circulation of freshwater through the engine jacket for cooling is accomplished by an engine-driven pump. The engine-jacket outlet water passes through a temperature regulating valve and jacket water cooler where a heat exchanger with salt water provides for cooling. The temperature-regulating valve bypasses a flow past the cooler sufficient to maintain the set freshwater temperature. Jacket water is discharged into the engine air coolers

which remove the heat of compression from the combustion air. Finally, the water returns to the jacket pump suction via the diesel lubricating oil cooler. During periods of shutdown, the engine is maintained in a ready status by a keep-warm water loop with a motor-driven pump circulating warm water through the engine jacket and bypassing the jacket water cooler.

6. Codag Start-Stop Reverse Sequences

As in the case of the COGAG plant, normal engine orders are executed by the base engine and boost-turbine throttles; but orders which require stopping or reversing the shaft rotation cannot be accomplished by this means alone, since the base diesel engine must be stopped and restarted in the opposite direction. This function is automatically carried out by the start-stop-reverse sequencer which controls the engine and turbine throttles, base and boost clutches, and the automatic diesel-engine starting system to carry out these orders without attention from operating personnel.

Operational situations in which frequent reversals are necessary, such as entering and leaving confined harbors, would require an excessively large air-storage capacity. To reduce this, a split-mode capability is provided. Here, the diesel engine is operated continuously in the astern direction and the gas turbine is operated in the ahead direction. The ordered direction is achieved by clutching

in the appropriate prime mover. These operations are also carried out automatically by the start-stop-reverse sequencer.

7. Codag Combined Plant Throttle-Control System

The philosophy and basic functions of this unit in the CODAG plant are identical to those of the COGAG plant.

The diesel base engine with its requirement for a two-speed reduction gear necessitates a somewhat different operational sequence. Operation to the maximum cruising speed of about 20 knots is possible on the base engine and low-speed gear connection with closed-loop speed control. For higher speeds, the system is shifted to the maneuvering mode, the boost turbine is started and/or clutched in, and the gear clutches are operated to shift from the low-to the high-speed connection of the base engine. In this range, closed-loop speed control is provided by the boost turbine with the base engine operated in an open-loop fuel-flow-governing mode. On speed reductions, the sequence is reversed.

## E. CONVENTIONAL STEAM PROPULSION PLANT

The plant consist of two conventional D.D. type boilers operating at 1200 psig, 950 F, and a single high-pressure/low-pressure cross-compounded turbine engine, together with the usual auxiliary machinery. The full power rating for the plant is about 35,000 shp.

Figure 36 is an overall control system diagram for the automated steam plant showing the interrelation-

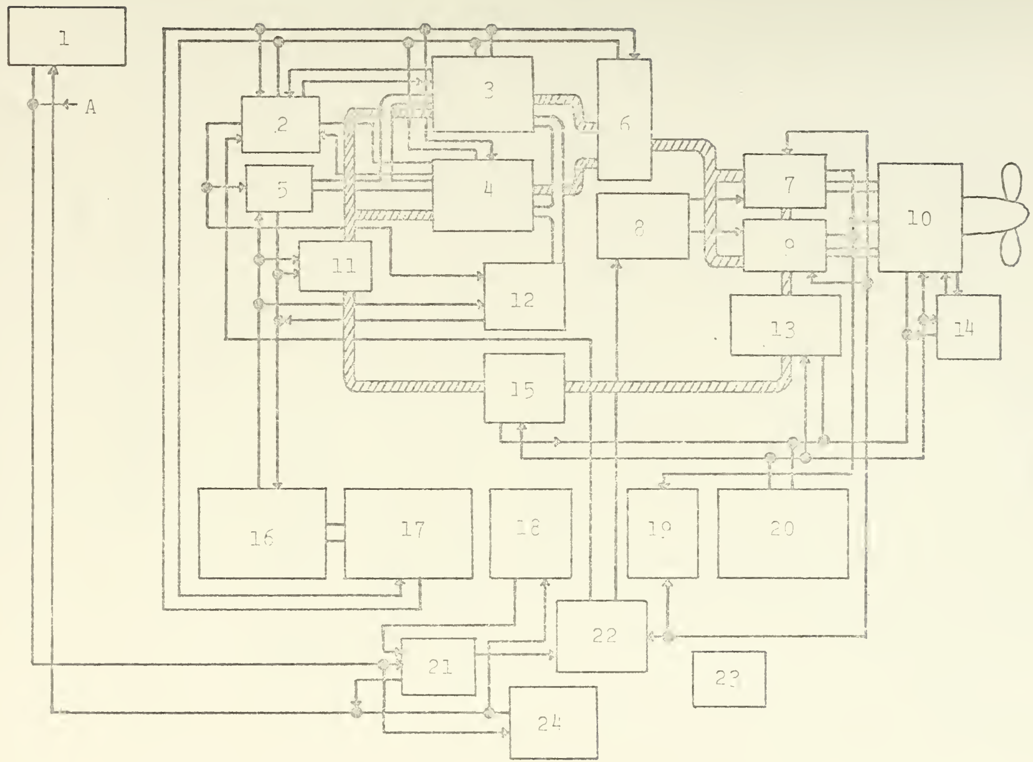



Figure 36. Automated Steam Propulsion Plant

Figure 36 (Continued)

Conventions

1 Bridge Propulsion Control
2 Boiler Control System
3 Boiler No. 1
4 Boiler No. 2
5 Fuel Supply System
6 Steam Distribution Controller
7 H.P. Turbine
8 Throttle Controller
9 L.P. Astern Turbine
10 Reduction Gear
11 Feed Water System and D.F.T.
12 Combustion Air Supply System
13 Main Condenser and Circulating System
14 Main LO System
15 Main Condensate System
16 Boiler Auxiliary Systems Instrumentation Control and Displays
17 Boiler Instrumentation Control and Displays
18 E.O.S. Propulsion Control
19 Engine Instrumentation and Displays
20 Engine Auxiliary Instrumentation Displays and Controls
21 Control Transfer Logic
22 Throttle Control System
23 Alarms and Displays
24 Plant Set-up Controls and Displays

A From Auxiliary Stations

ships of the major mechanical subsystems and their controllers.

The principal controlled quantity is shaft speed, regulated in a closed-loop mode. The plant operates as a conventional "boiler follower" in which the turbine throttle is operated as required to provide the desired shaft speed (subject to some limitations) and the boiler controls follow so as to maintain a constant steam pressure.

The general control design philosophy and operation is identical to that set forth for the COGAG plant, above. However, there are several substantial differences in its implementation due to the fundamental differences in the machinery components.

The most notable difference is the requirement for continuous regulation of the closed steam cycle, combined with the physical separation of the combustion process from the engine. This results in a requirement for controlling and coordinating a substantial number of discrete and relatively dispersed equipments.

A further significant difference in the case of the steam plant is the use of a portion of the steam output for a number of auxiliary power applications in contrast to gas turbine or diesel installations where separate auxiliary power sources are employed.

A major difference in the extent to which automation is applied should also be apparent. There is no provision

for automatic lighting off or of securing the boilers or engines in the steam plant. In the case of a diesel or gas turbine plant, automatic starting is considered necessary to take advantage of the rapid response inherent in the prime mover.

1. Main Engine and Control System

The main engine is typical of combatant ship practice, consisting of a crosscompounded high-pressure/low-pressure steam-turbine set driving the shaft through a locked-train, double-reduction gear. Speed control is by means of the ahead and astern throttles, of conventional type, but servo operated for remote control. The throttle control system provides completely automatic closed-loop speed and direction control of the main engine under all normal operating conditions, except lighting off and securing. The heart of this system is the "acceleration computer and throttle position signal generator" which translates the shaft rpm order input into an appropriate sequence of throttle orders to properly execute it. This control establishes the transient characteristics and the steady-state stability of the engine. The other components of the system are: the "speed-order cutback computer," which acts to close or partially close the throttle in case of certain serious malfunctions or casualties and imposes an upper speed signal limit during one-boiler operation; the "engine rollover controller," which automatically causes the engine to be rolled over periodically while in

a standby status; and the "mode setting controller," which operates the main condenser circulating-water system to handle astern and low-speed-ahead conditions without attention from operating personnel

2. Steam Generator and Control System

The automatic boiler controls include automatic feedwater regulators and automatic combustion controls. These could be pneumatic systems or solid state electronic controls.

Since the boilers will normally operate unmanned, automatic flame-failure detectors and electrical ignitors are required.

The automatic control system provides for completely automatic operation of the boilers and associated auxiliary systems for all maneuvering conditions in either one-or two-boiler operating models.

Automatic light-off and securing for the boilers is not provided, since it appears that the substantial increase in system complexity would not provide a commensurate increase in operating effectiveness.

The steam distribution control for both boilers includes the main and auxiliary steam stops which are power-actuated valves manually controlled from the EOS. Remote steam stop-operating controls (closing only) are provided at the damage control deck. Local manual operation is also available.

3. Combustion Air Supply System

The combustion air supply system for each boiler consists of two conventional steam-driven, horizontal-type forced draft blowers discharging into the outer casing of the boiler. A small motor-driven blower is installed for lighting off use. Normal operation of the steam units is fully automatic under control of the automatic combustion-control system.

4. Fuel Oil Service System

This system provides Navy special fuel oil (NSFO) to the boilers. It is of the conventional type with the addition of a few automated features and a substantial capability for remote operation. It consists of two main two-speed, motor-driven pumps, each capable of supplying the fuel requirements for full power operation of both boilers, and one two-speed, motor-driven pump of reduced capacity for port use. Two fuel-oil service tanks are provided, and in operation one pump is in use with the other pump lined up to the standby suction. Each pump discharges through steam heaters and a duplex strainer to the boiler front. Pressure regulation at the boiler front is accomplished automatically by pressure sensors at the boiler which operate bypass valves at the service-pump discharges. The service piping can be cross-connected at the pump discharges, at the heater outlets, and at the boiler fronts for isolation of various components in case of casualty. Pressure regulation to the burners is by

means of metering valves actuated by the automatic combustion-control system. Atomizing steam pressure is automatically controlled on the basis of fuel-oil manifold pressure. Emergency quick-closing valves are installed in the boiler front piping to each boiler and are operable locally and remotely from the damage control deck and the EOS. In normal operation, one pump, service tank, and heater are in use with the system cross-connected at the boiler front. For improved casualty resistance, split system operation can be employed. The automatic operation of this system includes the following functions:

1. Regulation of fuel-oil pressure at the burners by the automatic combustion-control system

2. Regulation of atomizing steam at the burners (indexed to fuel-oil pressure)

3. Regulation of fuel-oil supply pressure to the boilers

4. Control of fuel-oil temperatures from heaters (by local thermostatic control)

5. Regulation of the number of fuel-oil heater sections in use

6. Starting of the standby main pump in case of pressure loss

7. Cutoff of oil and steam to individual burners in the event of flame failure

8. Cross-connection at the boiler fronts in the event of pressure loss in one system during split operation.

5. Main Feed System and Deaerating Feed Tank

The principal mechanical components are three motor-driven feed booster pumps and three steam-turbine-driven main feed pumps. The booster pumps are arranged to take a suction on the deaerating feed tank (DFT) through a feedwater cooler. They discharge into a common suction line for the three main feed pumps. The main feed pumps discharge to the boilers through the feedwater regulator valves, which are automatically controlled by the feedwater regulator systems. Recirculation lines are provided at both the booster pump and the feed pump discharge. Two of the feed booster pumps are arranged to permit a cold suction on either one of two emergency feedwater tanks in the event of a deaeration feed tank (DFT) casualty. Two feed pumps and two booster pumps are adequate for operation at full power on both boilers. In normal operation, one or two pumps are in use with another feed pump warmed up and idling and one booster pump lined up to the cold suction. Operation of the main feed pumps is automatically controlled by a differential pressure regulator which controls the pump turbine throttles as necessary to maintain a fixed differential pressure across the feedwater regulator valves. Each feed pump is also provided with a recirculation control which recirculates sufficient discharge water back to the DFT to prevent pump damage at light boiler loads.

The EOS operator has the ability to start and place on line a main feed pump which has been properly lined up,

warmed up, and is idling. Remote manual starting of the feed booster pumps is also provided at the EOS.

From a casualty control standpoint, remote starting of major auxiliary machinery components is highly desirable. However, in the case of large steam-powered equipment, this presents a very complex problem.

The DFT is of unmodified conventional type. Water level control is automatic by means of the "floatless level control" which operates the valves required to dump excess condensate to the emergency feed tank and to take on makeup feed. Manual level control is also available if required.

6. Boiler and Boiler Auxiliaries Controller

This unit provides the control facilities as well as the instrumentation and displays required for operation of the boilers and their associated auxiliary machinery which together comprise the portion of the propulsion plant located in the fire-room. The boiler and boiler auxiliaries controls must be thoroughly integrated to achieve proper operation.

a. Boiler Control and Displays

(1) Automatic Combustion Control System and Diaplays. The automatic combustion-control system issues control signals to the fuel-oil metering valves, the forced draft blower dampers and throttles, and the atomizing steam control valves to provide the requisite fuel and air to maintain optimum combustion conditions at the proper firing rate. Feedback information from the control elements is

displayed for use by the operator in evaluating system performance and detecting improper operation.

(2) Automatic Feedwater Regulators and Displays. The automatic feedwater regulators maintain the proper steam-drum water level on all demand conditions. Appropriate feedback information from these controlled elements is also displayed.

(3) The Soot Blower Sequences and Timer. The soot blower sequencer and timer controls the soot blower steam root valves and the blower element drive motors as required for blowing tubes. The timer checks the proper progress of the sequence and interrupts it, if necessary, due to malfunctions. Displays are provided for the operator's use in determining the proper progress of the operation.

(4) The Boiler Casualty Alarm Display and Remote Manual Operating Station. The casualty alarm display and the manual control station summarizes and displays alarms relating to improper boiler operation and provides the capability for the operating personnel to assume remote manual control of any variable on either boiler from the automatic system when necessary for casualty control purposes, or to secure a boiler in an emergency. There is no provision for automatic boiler shut-down under any circumstances.

b. Boiler Auxiliaries Displays and Controls

This system is a loosely connected group of subassemblies which provide the instrumentation and displays

required for operation of the auxiliary machinery associated with the boiler. The several assemblies include:

(1) Feed booster pump automatic start-stop control and operating status display.

(2) The deaerating feed tank and water distribution system automatic controls and displays.

(3) The main feed pump controls including:

(a) Differential pressure-regulating system.

(b) Recirculating system control.

(c) Automatic Start-Stop Controls.

(4) Fuel Oil Service System automatic controls and displays.

(5) Forced Draft blower automatic controls and displays.

(6) Casualty-Control alarm displays and remote manual Control Station.

The casualty control and alarm display section provides a centralized indication of malfunctions in the boiler auxiliary systems. The manual control station provides the capability for operating personnel, to manually assume control of those systems elements, normally under automatic control in case of casualty, to select the operating/standby machinery, and to control the lineup of various piping systems.

c. Main Condensor and Circulating System

The main turbines exhaust to a water-cooled condenser of standard type. The condenser requires no

control in normal operation. The circulating system provides cooling water to the main condenser (and the main lubricating oil cooler). At moderate ahead speeds scoop injection is utilized. At low speeds and astern operation, a two-speed, motor-driven circulating pump is used. The pump and associated valves are automatically operated by the throttle control system to maintain proper circulation at all speeds and can also be operated manually from the EOS. Normally, no other system control is required, but under unusual operating conditions, particularly very cold water, the overboard discharge valve is manually operable to throttle the water flow and prevent excessive condensate cooling. For emergency dewatering, a bilge suction is provided for the circulating pump under manual control.

d. Main Condensate System

This system removes condensate from the condenser hot well and delivers it to the DFT after using it as coolant in several heat exchangers. Two motor-driven condensate pumps each capable of operation at full plant load are installed. Operation of these pumps and their associated valves is manually controlled from the EOS. The discharge condensate from the pumps is employed for cooling in the condensers associated with the main air ejectors. A thermostatic control provides for circulation from the air ejector outlet back to the condenser when necessary. Otherwise, condensate is piped to the deaerating section of the DFT through the feed booster pump suction precooler. The Fresh

Water Drain Collecting Tank (FWDC), accumulates water from miscellaneous steam and fresh water drains, and makeup feed from the reserve feedwater tanks, discharging it into the condensate system by two automatically controlled and sequenced motor-driven pumps through valves operated by the DFT level control. Excessive water in the FWDCT is handled by a float-operated valve in a vacuum drag line to the main condenser. The condensate system is self-regulating during normal operation and requires only monitoring and alarms to detect abnormal conditions.

e. Reduction- ear and Main Lubricating Oil System

The main turbines drive the propeller shaft through a conventional locked-train double-reduction gear. The main lubricating oil system provides lubrication to the high and low-pressure turbines, the reduction gear, and the main thrust bearing. These are not significantly different from the units in the COGAG plant.

f. Engine and Engine Auxiliaries Controller

This unit provides the control facilities as well as the instrumentation and displays required for operation of the main engine and associated auxiliary machinery which together make up the portion of the propulsion plant located in the engine room.

Primary control of the main engine is exercised by the throttle control system. This unit accepts, through the control transfer logic, the remote input engine orders and operates the throttles to ensure proper execution of

these orders. There are two primary remote control points, one at the EOS and one at the conning station.

The engine auxiliaries controllers provide the instrumentation displays, alarms, and controls necessary for operation of the condenser, condensate and main circulating systems, and the main lubricating oil system. The automatic functions provided include:

(1) Starting/stopping of main circulating pump and associated valves as required by ship's speed order.

(2) Starting of the standby lubricating oil pump on low pressure.

(3) Starting of the emergency lubricating oil pump on lower pressure.

(4) Regulation of lubricating oil temperature from cooler.

(5) Recirculation of condensate from air ejector outlet as necessary.

g. Miscellaneous Systems

By the nature of the steam propulsion plant, there are other systems which provide support or essential services for the propulsion installation which are not found in the previous propulsion plants. Some of these systems are:

(1) Auxiliary Exhaust System. Steam driven auxiliaries which operate in noncondensing modes and some other equipment exhausts into the auxiliary exhaust system which is used for feed heating and deareation, as a power

source for the distilling plant and for other miscellaneous purposes. This system is intended to operate at a constant pressure, with excessive steam dumped to a main or auxiliary condenser and inadequate steam augmented by reducing stations from the 1200 psi auxiliary steam line. System pressure is displayed and alarmed at the EOS.

(2) Gland Seal Steam System. Gland sealing steam for the main turbines is provided by an automatic steam seal regulator operating from 150 psi auxiliary steam. Seal steam pressure is indicated and alarmed at the EOS.

(3) Main and Auxiliary Gland Exhaust System. These system are self-regulating and require no instrumentation or alarming.

(4) Auxiliary Steam Systems. These include the 1200 psi auxiliary system and the 150 psi auxiliary system. These are employed as power sources to various other pieces of auxiliary machinery. The 1200 psi system is supplied directly from the boilers through the internal desuperheaters, and the 150 psi system is supplied through automatic reducing valves from the 1200 psi system. These systems require no control or instrumentation beyond pressure indications at the EOS.

(5) Fuel Oil Filling, Transfer and Stripping Systems. The operation of these systems is manual, since gains in effectiveness from their automation would be only marginal. Consequently, no instrumentation or control facilities are required.

(6) Saltwater-Service System. Saltwater service is required for a number of functions, principally machinery cooling. This is provided from the fire main system. System operating pressure is displayed at the EOS. A pressure-failure alarm is installed.

## F. BASIC POINTS FOR AUTOMATIC PLANT DESIGN

### 1. Discussion of Design Approach for Specific Systems

This second section will review certain recurrent problems in the control of naval machinery and the possible methods of solution.

#### a. Steam Generator Water Level Control

Boiler feedwater regulators were among the earliest automatic control systems to find application in marine steam propulsion plants. Present day control systems are an evolutionary descendant of the simple single-element thermo-hydraulic devices employed on some World War II vessels. In general, these systems did not provide satisfactory performance (in terms of current requirements) and are now obsolete. The addition of flow elements and the use of instrument type controllers constitute the major advances in this area. The design objective for these systems is, of course, the combination of simplicity, reliability, and compliance with performance specifications. The selection of the type of system to be employed, one, two or three element feedwater regulator depends on the characteristics of the steam generator and the detailed

requirements of the performance specifications. The choice can be readily determined if the following questions can be answered:

(1) What is the allowable transient deviation in water level relative to the magnitude of inherit "shrink" and "swell" in the boiler?

(2) Are there specifications concerning the transient response characteristics of the controlled feedwater flow rate?

(3) Are there specification requirements concerning the steady-state stability of the level, flow, and control impulses? If the replies to the latter two questions are negative and the allowable deviation in level is small, a single-element regulator is indicated. Although there are some exceptions, the multi-element regulator generally promotes stability with some sacrifice in water level excursion.

It is important to consider the desired characteristic of feedwater flow transient response in terms of its influence on the performance of the main feed pumps, deaerating feed tank, and auxiliary exhaust system.

Without consideration of the economics involved, according to the studies of the naval boiler and turbine laboratory (Philadelphia Naval Shipyard) the optimum control system for applications to processes with unknown dynamic characteristics is one in which the difference between metered steam flow and feedwater flow (measured by

extraction of the square root of differential pressures created by flows through metering elements) is applied as a set-point signal to a controller whose feedback loop is closed through the boiler water level. Filtering or damping can be applied as required. The feedwater flow control valve should be designed for a linear low variation as a function of control pressure and should be equipped with a force-balance positioner. This system offers the maximum versatility (except for those designs offering automatic boiler blowdown compensation) in terms of programmability. The relative sensitivity of the controlled feedwater flow rate to disturbances in load, level, feedwater header pressure, and feedwater flow can be adjusted over a wide range. Experience with this system arrangement on many different types of marine boilers has indicated a nearly universal capability to provide optimum water level control.

The single and double element regulators, on the other hand, are more restricted in application in that they are more dependent on both the static and dynamic characteristics of the boilers and the feedwater control valves to which they are applied. Furthermore, the tendency to augment interactions with the main feed pump control system, thereby producing loop instability, is more difficult to eliminate by adjustment. The two-element system depends on a linear control valve characteristic in order

to maintain the desired water level at all steaming loads throughout the range.

b. Forced Draft Blower Control

The problems incurred in obtaining satisfactory closed-loop control of main forced draft blowers have probably caused more practical difficulties in both new construction and operating vessels than in any other single aspect of plant automation. This situation has resulted from the combined inherent dynamic non-linearities of rotating machinery, and the static nonlinearities, dead-band, and backlash, that have been unwittingly designed into many of the units produced for the Navy.

The dynamic nonlinearities of main forced draft blowers arise as a consequence of the angular corollary to Newton's Second Law:

$$\frac{d\omega}{dt} = \frac{1}{J}\left(T_a - T_r\right)$$

where:

$\omega$ = angular velocity, rad/sec

$J$ = polar moment of inertia, ft lb sec

$T_a$ = driving torque, lb ft

$T_r$ = retarding torque, lb ft

The torque terms are both nonlinear functions of other parameters of the system. Substitution of these and other parameters leads to the relationship

$$\frac{dQ}{dt} = \frac{\overline{Q}}{\overline{\omega}J}\left[\overline{T}\left(\frac{G}{\overline{Q}}\right)^{2/3} - \overline{T}\left(\frac{Q}{\overline{Q}}\right)^{2}\right]$$

where:

$Q$ = blower air delivery

$T$ = torque

$G$ = turbine steam consumption

and the bar designates the value of the variable at the design rating.

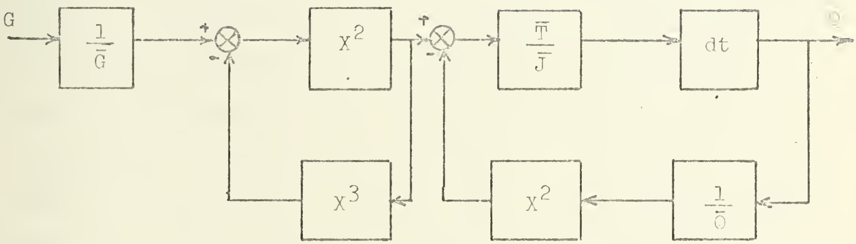
Linearizing this differential equation by the perturbation method produces a "transfer function" of the form:

$$\frac{Q}{G} = K \left(\frac{1}{1 + Ts}\right)$$

This first order lag is characterized by a decreasing value of K (open-loop gain) and a decreasing value of T (response time constant) as blower speed is increased. Experimental work has confirmed the validity of this analysis for several different blower designs.

The blower itself can thus be thought of as an ordinary differential equation having variable coefficients, or as a nonlinear feedback system of the type diagrammed in Figure 37.

In addition to the nonlinear functions inherent in this loop, the blower steam flow regulating devices found in almost all applications produce nonlinear steam flow variations as a function of control signal input. Thus, the overall nonlinear loop sensitivity is further compounded. This means that the systems currently in use in Naval applications exhibit considerable less than the optimum combination of response and stability. The general




Q = blower air delivery

G = turbine steam consumption

T = torque

J = polar moment of inertia (ft,lbs,sec)

The bar designates the value of the variable at the design rating.

Figure 37. Schematic Representation of a Forced Draft Blower

approach to this problem has been to "tune" the system for best response at the condition of maximum open-loop sensitivity (of the boiler and blowers), and to accept whatever conditions occur at other loads. The result of this method is usually a compromise, wherein the air flow control loop exhibits a slight instability at the full power condition; and, because of excessive gain and phase margin, a slow and sluggish transient response at low loads. This problem can be avoided by establishing design specifications which provide good dynamic design because although the cost of the equipment furnished may increase at a rate that offsets the performance gains, improved smoke control reduces boiler maintanance and improves plant availability and efficiency.

To improve the standards for combustion airflow, it is necessary to specify transient response under conditions of both anticipated loading and specified input, which should provide design guidance which will produce the desired performance. Inspection of Figure 37 illustrates the type of approach that might be taken. The general requirements are that the gain of the feedback loop be minimum, and that the relationship of steam flow to control signal be such as to produce steady-state linearity between air delivery and control signal. In the forward loop, the quantities $\overline{T}/\overline{G}$ and 1/J should be as large as is feasible. The $\overline{T}/\overline{G}$ term is established by the ratio of turbine torque to steam flow and is thus dependent on the design and rating of the auxiliary driving turbine. The torque-flow relationship

should be as steep as possible; this indicates the need for high efficiency in the nozzle and blade system. The moment of inertia of the rotor assembly should be minimal indicating the need for lightweight materials, small parts and compact assembly. Undoubtedly, the bladed turbine design offers advantages over the "bucket-wheel" design in all of these areas. The gain of the feedback loop is a function of the static pressure rise across the blower and is, therefore, dictated by boiler requirements. It should be remembered, however, that reducing gain in the feedback loop not only increases the rate of speed increase but also reduces the rate of speed decrease. A high speed blower combined with a low draft boiler is, therefore, not calculated to preclude the formation of white smoke when load is reduced. The feedback arrangement serves to illustrate the fact that the chief virtue of combustion control dampers is not in their effects on air delivery caused by system resistance changes on the blower map, but in their effects on the feedback loop gain, increasing it as power is reduced and decreasing it as power is raised.

Satisfactory smoke control can be achieved without the use of combustion control dampers provided that the time constant T of the forced draft blower, evaluated at the all-blowers-in-use cruising boiler load is six seconds or less, where the cruising load occurs at 50% of boiler rated full power or less. Since the time constant is equivalent to the ratio of the polar moment of inertia to

the rate of change of torque with respect to angular velocity, it can be evaluated from the design.

Referring again to Figure 37, it can be seen that the effect of the second order feedback term around the integrator is to reduce the loop gain at high loads. It is recalled that the steady-state speed varies with the cube root of the steam flow, thus indicating the need for nonlinear valve and actuator characteristics that cause flow to vary with the cube of lift. The simplest approach to design for linear forced draft blower speed control is to specify a single throttling type control valve characterized by plug design for the desired relationship or equipped with a characterizing cam operated positioner. The initial cost of the single valve arrangement is lower than the multiple nozzle/poppet valve arrangement used to get the best possible plant heat rate because we can consider that economy is not sacrificed at high loads (particularly if a separate overload nozzle valve is provided); the maintenance costs are greatly reduced; and blower reliability is improved.

Other factors outside the province of the blower designer also contribute effects to air flow control loop performance. The nonlinear characteristics of the steam generator itself contribute to the general decrease in gain margin with increasing load. The method of air flow measurement and feedback is open to choice; in those applications wherein the total air flow is metered, the

closed-loop sensitivity of the air flow control system is independent of the number of burners in use in the boiler. When air flow is metered on a per burner basis, the loop gain increases with the number of burners in operation. In either case, the loop gain varies directly with the number of operating blowers.

In general, the method of closing the air flow loop employs a proportional plus integral controller whose output regulates air flow and, in some cases, combustion control damper position. Because of the many nonlinearities and integrations present in most systems, these control loops are usually characterized by extremely low reset corner frequencies. This results in the undesirable effect of a prolonged time required to complete the response of steam pressure following an increase in boiler load.

c. Fuel Oil System Control

The method of fuel control for any given application depends both on the type of fuel burning equipment to be employed and on the general philosophy on which the entire boiler control system is to be based. The basic types of wide range fuel burners used in modern marine power plants include the return flow mechanical atomizing and the steam assisted atomizing type. In either case, a choice must be made whether to regulate the total fuel rate to the boiler or the fuel rate per burner. In making the selection, some consideration should be given to the relative

effects of the two methods on the combustion air flow system characteristics.

If the total fuel philosophy is adopted, the control functions are a straightforward adaptation of the principles of measurement and feedback control. The fuel oil burners themselves are not characterized by dynamic properties that create difficulties in the closed loop. On the other hand, if the "per burner" fuel system is selected, flow is controlled by means of an inner loop fuel pressure control system, and the flow demand signal must be reshaped to form a pressure demand signal. In this case, whatever nonlinearity exists in the fuel atomizers must be compensated by a programmed inverse in the control system. Unfortunately, these nonlinearities do not remain constant; they are functions of sprayer plate wear, fuel temperature, and fuel supply pressure. The principal difficulties with systems of this type are encountered when the fuel oil sprayer plate characteristics undergo radical changes in slope of the pressure/firing rate relationship. The extent of these variations varies widely from one atomizer design to another, and from one sprayer plate to another. In general, the inverse nonlinearity of this function must be permanently programmed into the system. As the slope variations of the sprayer plates become more and more extreme the required variations in the closed-loop gain of the nonlinear function generator become correspondingly wider. In view of the fact that all function generators currently

offered for this application are some form of closed-loop feedback mechanism, it is easy to see why stability of the function generator itself might become critical in certain portions of its range. Thus, the choice of the type of fuel oil control system to be selected for any given application can be seen to depend on other than purely arbitrary considerations. As a general rule, it has been suggested that the ratio of the maximum slope of the fuel pressure vs. fuel rate curve, to the minimum slope (within the normal operating range) be used as criteria for the selection of basic system design. If, for the sprayer plate chosen for use with the steam generator, this ratio exceeds about ten to one, the characterized fuel pressure control system with force-balance fuel control valve should not be used. The limit is, of course, established by the gain and phase margin of the particular function generator chosen for a given design, but the ten to one limitation should prove adequate for most applications.

In the case of steam-assisted atomization, the characteristic curves of typical steam atomizers are usually sufficiently linear to permit the use of either type of control system. The principal difficulties encountered in the design of control systems for such applications are the extremely low fuel pressures involved and the difficulties in dynamic control of the desired relationship between steam pressure and fuel pressure. In order to provide the simplest, most reliable, most economical control system compatible

with specified performance, the simplest possible linear mathematical relationship between fuel pressure and steam pressure should be adopted. The use of feedback loops that meter steam and fuel pressures should be avoided wherever possible. The automatic reset mode should be avoided at all costs, since this introduces phase shift at low frequencies and produces dynamic errors in maintaining the desired fuel/steam relationship. This can result in serious combustion difficulties when maneuvering at low rates. In view of the desirability of an open-loop or even feed-forward approach to atomizing steam control system philosophy appears to offer some advantages. On the other hand, the metering systems avoid the problems inherent in this type of fuel control system and are a more straightforward application of linear feedback systems design. Each case must be considered on its own merits, not the least of which is the simplicity and lower cost of the characterizing system. The important point is that the choice is not a purely arbitrary one based on cost or similar considerations alone. A basic design study to evaluate the merits of the two philosophies for any particular application prior to preparation of detail specifications is required.

d. Boiler Steam Pressure Control

Analysis of the dynamics characteristics of main propulsion steam generators, has disclosed that these dynamics are essentially nonlinear. Furthermore, because of the thermal storage capacity available, the steam pressure

variation of a boiler behaves fundamentally as the time integral of the heat output/heat input difference. The nonlinearities produce variations in the integration coefficient. These observations suggest that: (a) control system phase margin will be difficult to obtain, and (b) no linear feedback control system can produce optimum response at all conditions.

There are two fundamental types of steam pressure control systems found in current Navy combustion controls. Both of these systems use linear components, and both employ a steam pressure transmitter to close a feedback loop. One of these systems uses the automatic reset mode to eliminate offset errors at steady-state loads. The other system employs an impulse proportional to boiler load as a substitute for reset feedback. By calibrating the feedback signals of the air flow and fuel flow inner loops, the proportional offset and low frequency phase lag are both eliminated. This arrangement, known as the "two-element-master combustion control system" is characterized by a generally better dynamic transient response and greater stability at steady state. Unfortunately, this system requires that all feedback signals be proportional to total load, not load per burner, which thereby limits the application to those systems in which fuel rate is metered, or to those systems in which the number of operating burners remains constant under all steaming conditions. The improved performance of this system in comparison with the

reset type system should, therefore, be considered in judging the relative merits of the flow metering fuel oil control system.

In consideration of the nonlinear aspects of steam generator dynamics, it became apparent that an ideal control system is one which compensates for variations in the open-loop characteristics in such a manner that the closed-loop gain and phase are maintained in the desired relationship to each other. Analytical derivation of the nonlinear differential equation for the response of steam drum pressure to changes in the steam generator heat balance indicates that the open-loop gain increases with load.

A nonlinear control system design, based on the concept of gain variation in the compensation in order to preserve essentially constant gain in the closed-loop was conceived and installed for evaluation on the DLG-9 Class test boiler. Results of maneuvering tests under extreme transient conditions indicated that the maneuverability and response of the system were vastly superior to those of the unmodified control system. In addition to improved response in the steam pressure control loop, it was found that nearly twice as much gain could be accomodated with good steady-state stability in the air flow loop.

The system conceived for providing the required gain correction is shown diagrammatically in Figure 38. It utilized a Bailey Meter Company Type AR8X31A, Model A, computing relay in place of the conventional master relay of

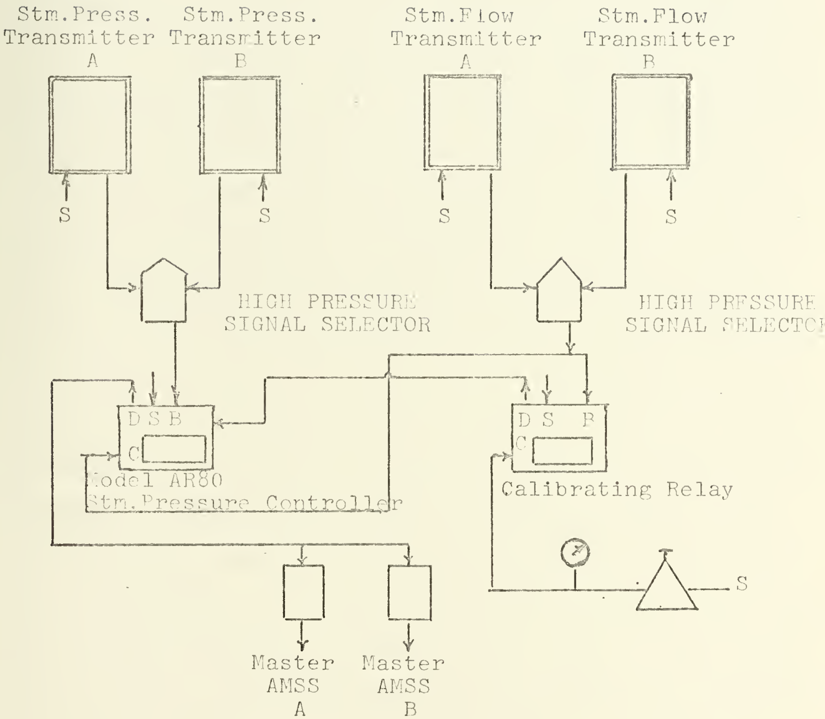



Figure 38. Arrangement of Steam Pressure Control System Using Non-Linear Controller

the two-element system. The unit was calibrated to vary from 100% proportion al band to 10% proportional band as the pressure in the driving bellows was veried from 3 to 27 psig. This signal, supplied to the "L" connection of the relay, was received from a calibrating relay suitable for applying gain and bias to the primary steam flow signal.

Using this system, it was possible to produce transient load changes within specification steam pressure deviation with steam flow changes several times faster than specification. Steam flow increases from a standby load of 16,000 pounds of steam per hour to 90% of full power (152,000 pounds per hour) in nine seconds produced a drop of only 110 psi at the superheater outlet. Increases in load in the specified time, 25 seconds, produced drops of only 60 psi in contrast to the 100 psi deviation normally occurring with the standard system. This improvement occurs primarily because of the considerably more rapid dynamic response of the combustion air flow loop than in the conventional system; as a result, the pressure deviation during load reduction was only slightly better than normal (30 psi in lieu of 35), but maneuvers could be executed more rapidly with less tendency to produce white smoke. During a load decrease from 90% of full power to standby with all burners in ten seconds, a brief puff of while smoke, sustained for about five seconds, was experienced. During normal load decreases on DLG-9 Class ships, white smoke is invariably

produced; standard maneuvers with the modified system produced no smoke whatsoever.

It should be remembered that this type of control action is not limited to the two-element steam pressure controller, although it performs to best advantage in this case. It should also be noted that its function is not to compensate for the nonlinearity introduced by super heater pressure drop in applications where the superheater outlet is the control point, but rather to compensate for nonlinearity within the steam generating portion of the cycle.

e. Fuel Oil Service Pump Governors

Control of discharge pressure at steam turbine driven fuel oil pumps has been traditionally accomplished with single-element direct actuated control valves with a hydraulic feedback line. In many applications, these devices have proved eminently successful; in many others, the identical designs have always shown gross instability. This instability is believed to be caused by several factors or a combination thereof:

(1) interaction of pump control and combustion control.

(2) improper location of fuel pressure sensing connection.

(3) poor quality control of the regulating valve manufacturer.

These problems seem to be unpredictable; and in many cases, little can be done to correct them. It would appear that a better approach to control system design for fuel oil service pumps is to consider the pump an integral part of the total fuel oil control system, and to design the controls with this in mind. Thus, the regulating valve could be programmed to respond directly to flow variations indicated by the sum of the metered and transmitted fuel supply rates to the boilers with fuel pressure error applied as a corrective override signal. Fail-safe operation can be provided by a backup system designed to revert to single element direct action control in the event of loss of pneumatic air supply.

A typical Fuel Oil Service Pump Control System is indicated in Figure 39.

f. Main Feed Pump Control

Two control functions are performed by the general group of control systems applied to main boiler feedwater pumps. These include regulation of differential pressure across the boiler feedwater control valves, and regulation of by-pass flow around the pumps when operating at light loads. Existing systems in current naval use perform quite satisfactorily with the possible exception of occasional rather strong interactions between feedwater regulators and feed pump differential pressure controls. Once again, these situations arise out of a failure to approach the control design from a "system" point of view.

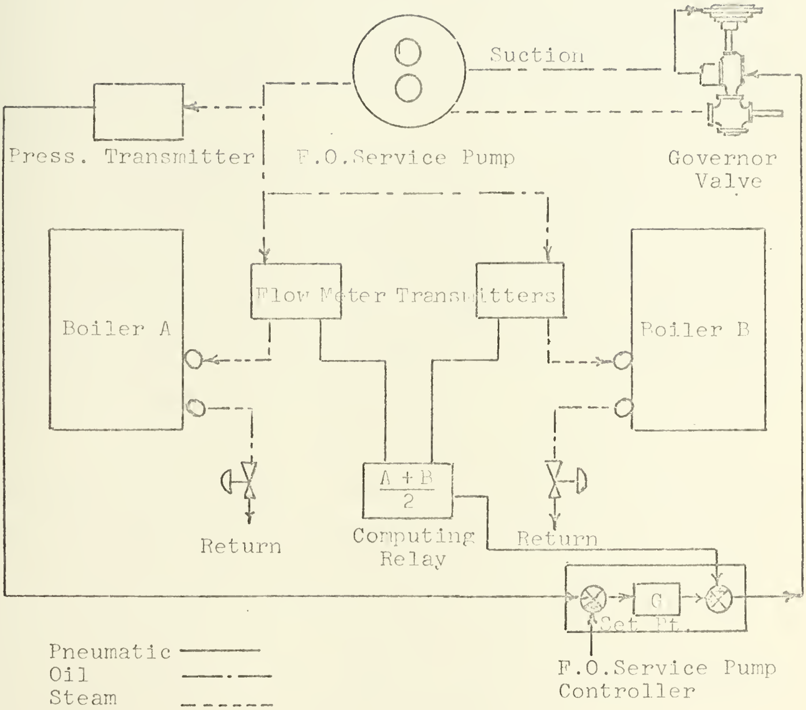



Figure 39. Fuel Oil Service Pump Control System

The actual exchange of information between pump and feedwater control systems is probably impractical in view of the many combinations of pumps and boilers that might be operating at any given time. As the integrated plant concept finds its way into actual applications, however, the use of logic blocks will permit the application of integrated techniques to the combined problem of pump control and boiler feedwater control.

g. Auxiliary Steam Systems

Steam pressures in auxiliary systems are regulated by pressure reducing or back-pressure regulating valves in existing steam plants of naval ships. An unestable pressure control system is actually a very common problem in automated installations.

The usual cause of these difficulties in "hunting" of the pressure reducing valves that supply steam from the high pressure desuperheated steam system into the lower pressure auxiliary steam system; in some cases, such hunting is being forced by unstable regulators supplied from the auxiliary system, such as pump governors, augmenting systems, etc. The reducing valves that service auxiliary systems are usually of the force-balance spring-loaded type, with an external sensing line. One of the chief causes of instability can be traced to selection of the sensing point on the downstream piping for the feedback connection. The cardinal rule to be observed is to locate this connection at a point in the system where the

vena contracta effects at the valve discharge have dissipated, and in a sufficiently large piping section that steam velocities are low. On the other hand, the sensing point should not be so far removed from the controller that dead time is introduced into the feedback loop.

Control of auxiliary exhaust pressure has also been found to be almost universally unsatisfactory in most combatant vessels. The basic cause of this performance appears to be attributable to underdesign of the auxiliary exhaust spill control valves to the main condensers. The capacity of these valves is usually predicated on exhaust steam flow rates computed from ship heat balance data, which in turn is based on estimated auxiliary turbine water rates. These figures are predicated on optimum plant line-up for maximum economy at each underway condition, a condition which does not always exist under actual steaming conditions. Furthermore, the allowance margin in sizing these valves, while probably sufficient to accommodate deterioration in the steady-state performance of the plant. does not provide sufficient excess capacity to accommodate transient conditions.

Consideration should be given to designs in which the augmenting and unloading exhaust steam control valves are actuated by a split range signal from the same controller. The control point should be located near the deaerating feed heaters, and a single control system provided that treats the problem of exhaust pressure control

as one requiring system design and analysis, not valve selection from a catalog.

It should be noted that the sizing of condensate pumps and even of feedwater heaters is subject to the same effects of boiler volumetric displacement as the sizing of exhaust valves. The causes of loss of water in condensate systems are almost always traceable to failure of the system to deliver the flow demanded by the controls, seldom to failure of the controllers themselves.

2. <u>Realizability of Digital Filters Operating in a Closed Loop Feedback Configuration</u>

Automatic control of conventional Navy ships to date has not utilized the great versatility and high reliability of the digital computer. In part, this is due to the fact that digital control is a relatively new concept, the potential of which is only now becoming fully realized. A more practical consideration is the problem of adding a digital computer to the already excessive number of special-purpose equipments on board today's operational ships. However, the fully integrated naval ship design makes possible the consideration of digital computer control of modern and future steam and gas propulsion systems.

The Naval Boiler and Turbine Laboratory (NBTL) in Philadelphia has developed an analog simulation of a DLG-9 class automatic combustion control system. The simulation is based upon the results of a test program to determine the open-loop dynamic characteristics of DLG-9 main boilers, forced draft blowers, and other control components. Load

dependent transfer functions comprise the boiler simulation while the forced draft blowers and actuators are simulated by a nonlinear differential equation. In each case, linearized transfer functions at a particular operating load level were used to prove local stability of the system. The remainder of the control components was considered to be linear throughout their operating range.

The combustion control system is shown in Figure 40. The superheater outlet pressure and drum water level are regulated to the desired set points by individual control loops. There is interaction or coupling between these two loops which is shown in this figure as $(\partial P/\partial G\omega)$ and $(\partial L/\partial Gf)$. In addition to the pressure and water level loops, there are two minor loops for control of oil flow and air flow. The air flow loop is seen to assume two positions depending upon the sign of the air flow error signal. For increasing steam flow, the air loop is in series with and provides the demand signal for the fuel loop. For decreasing steam flow, the air loop operates in parallel with the fuel loop. In this latter case, the pressure control loop operates independently of the air control loop. The system must be stable in both configurations to guarantee asymptotic stability.

In order to provide digital control for the system various continuous signals were sampled and transmitted to a digital computer via A/D conversion equipment. Figure 41 illustrates the sampled system with the functions of the

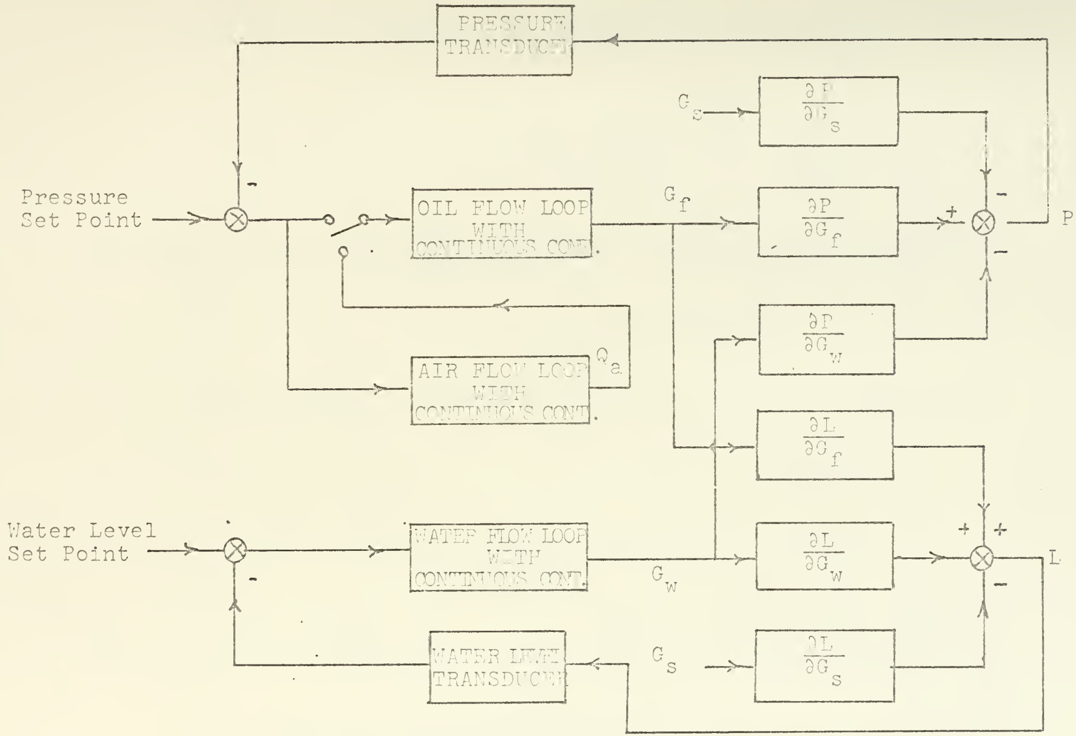
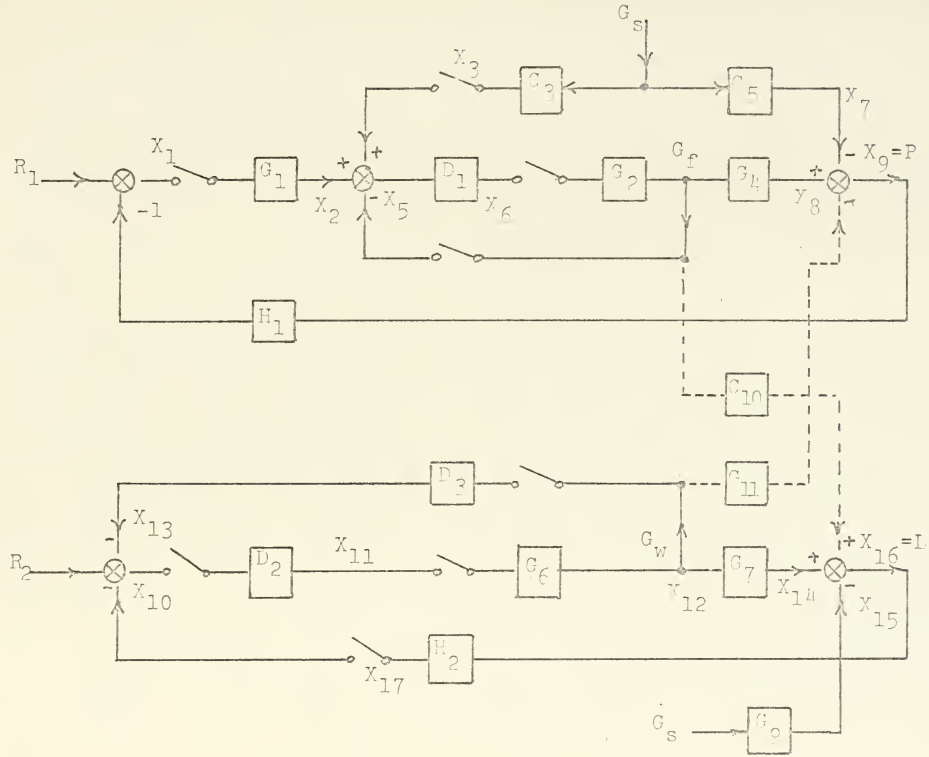



Figure 40. Block Diagram of Combustion Continuous Control System

Figure 41. Sampled Block Diagram for Decreasing $G_s$



continuous filters and analog logic being accomplished within the computer. In theory, the sampled signal flow diagram in Figure 42 can be utilized to obtain a complete analytical description of the system at the sampling instants. However, due to the wide range nonlinearities of the plant, the resulting equations will not be solvable by linear techniques. While these nonlinearities present a major impediment in a complete analytical treatment of the system, they present only slight difficulties in a design based upon the stability of the system's characteristic equation. From the linear dynamics of the plant at each load implies a system which is globally stable. The linear dynamics are known at the 28% and 77% maximum load conditions and are assumed to vary continuously between these levels. While the design of linear controllers to provide stability at the 28% and 77% levels does not necessarily guarantee global stability, it does give considerable insight into the design problem.

Using Mason's Gain formula the systems characteristic equation becomes:

$$\Delta = \{1 + G_1^* D_1^* (H_1G_4G_2)^* + D_1^* G_2^*\}\{1 + D_2^* (H_2G_7G_6)^* + G_6^* D_2^* D_3^*\}$$
$$+ G_1^* D_1^* D_2^* (H_2G_{10}G_2)^* (H_1G_{11}G_6)^*$$

where:

$G^*$ and $H^*$ represent the transfer function of G(s) and H(s)

$D^*$ represent the digital controllers.

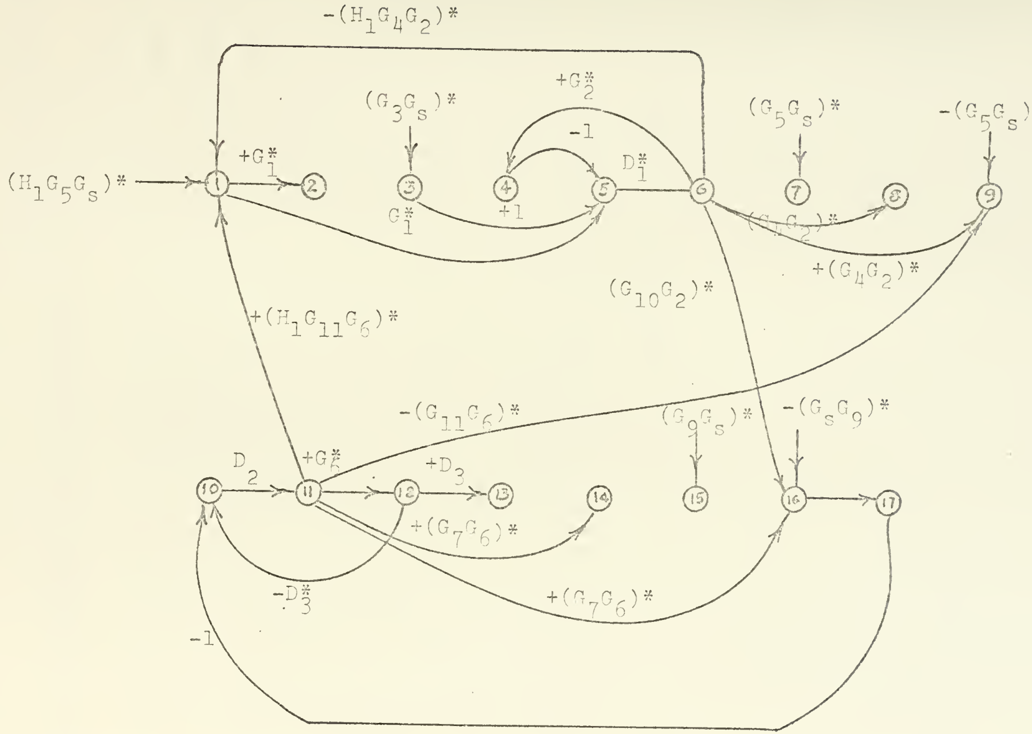



Figure 42. Sampled Signal Flow Diagram for Decreasing $G_s$

The characteristic equation is seen to be the product of the characteristic equations of the two major loops plus a term due to interaction between them. If the last term is negligible, then system stability can be guaranteed by the stability of each loop and single loop design in the z plane can be utilized to develop the various controllers.

The z transform is defined as:

$$z \equiv e^{TS}$$

$$s = \frac{1}{T} \ln z$$

$$z\{f(nt)\} = \sum_{n=0}^{\infty} C_n a^{-n}$$

The digital controllers developed for this system must necessarily be physically realizable. An open loop digital filter, D'(z), will be realizable if the present output is composed only of past values of the output in addition to present and past values of the input. However, there are cases when digital filters, realizable in an open loop configuration, can not be realized in a closed-loop feedback control system. An additional requirement for realizability of closed loop digital controllers is that the input to D'(z) should not respond instantaneously to the output of D'(z).

The controller would possess this additional requirement of realizability if

$$\lim_{z \to \infty} D'(z)\, G.H.(z) = 0$$

If the above is not true the effective controller D(z) behaves as $D(z) = Z^{-1} D'(z)$.

In the case of the DLGg boiler, the analysis of the pressure loop in the z plane was considered, but since the evaluation of the final value theorem to its transfer function is different from zero the control was unrealizable and $D'(z) = Z^{-1} D_1'(z)$. Using Root Locus techniques (Figure 43) suitable conpensation was verified at a 77%, a 28% load level using the values $D_1(z) = D_{1A}(z) D_{1B}$

$$D_{1A}(z) = \frac{3.21(z - 0.72)}{z(z-1)}$$

$$D_{1B}(z) = \frac{0.20z}{z - 0.80}$$

G. CONCLUSIONS

1. A review of the propulsion plants described in Section V-C,D and E shows the following important differences affecting the automation of such plants:

The diesel engine lubricating oil system is more complex in its control aspect than that for a gas turbine. In addition the diesel engine requires a cooling system consisting of fresh water and salt water. Gas turbines do not have a cooling system. Automatic start/shut-down is of comparable complexity for both diesel and gas turbines. For diesel this system involves a compressed air system, engine cam positioning, fuel rack position, and a number of interlocks and alarms. For gas turbines it involves a

$$G_1(Z)H_1G_4G_2(Z)+G_2(Z) = \frac{(Z-0.815+J0.189)(Z-0.815-J0.189)}{(Z-1)(Z-0.668)}$$

$$D_{1A}(Z) = \frac{3.21(Z-0.72)}{Z(Z-1)}$$

$$D_{1B}(Z) = \frac{0.20\ Z}{Z-0.8}$$

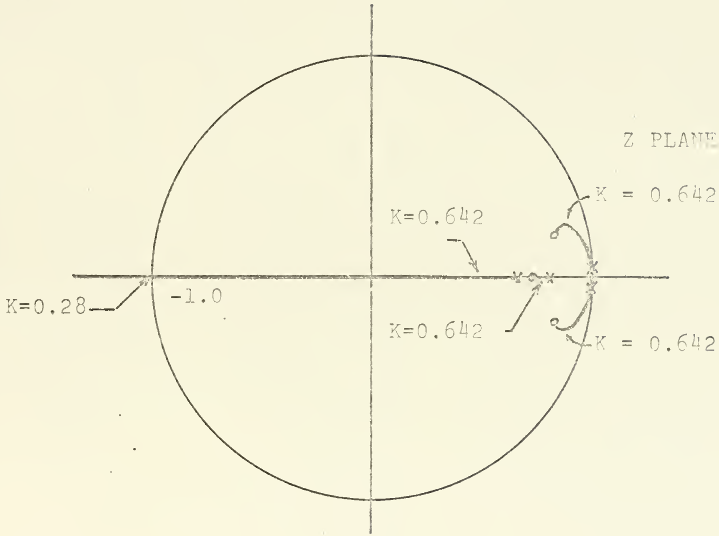



Figure 43. Compensated Root Locus for Pressure Loop with Decreasing $G_s$

hydraulic start motor, fuel and ignition control, and a number of interlocks and alarms.

The steam plant is notably different from the internal combustion engine types in that it alone employs a closed thermodynamic cycle, requires fuel heating, externally powered and controlled combustion air supply, uses steam turbine driven auxiliary machinery rather than electric motor driven ones. In contrast with diesel and gas turbines, direction control of the steam plant is very simple. From the operational point of view, the most significant variation between the plants is the absence of automatic starting and shut down in the steam plant.

Given the description of the systems, the main differences discussed above, in general terms but not in a very strict way, we can say that the following would be the order of decreasing control complexity of the propulsion plants considered before:

1. Steam
2. Cogag Electric (R.G.T. base)
3. Cogag Electric (Single Cycle Base)
4. Codag
5. Cogag (reversing gear).

2. The Automatic Control Design of the propulsion plant can not try to optimize only the subloops, but the total behavior of the plant as a whole.

3. The application of classical methods such as the root locus and Z transform in the design of automated Propulsion Systems, must be considered. Digital Computers give to the Naval Engineers the capability for convenient simulation of their systems and their corresponding control systems. The use of simulation programs such as the D.S.L. had been proved to give outstanding results and a great saving of time in simulation of Naval Propulsion Plants.

## VI. FINAL COMMENTS

After having defined the criteria of Naval Ship Control Systems and having had the corresponding investigation concerning the characteristics and mathematical models, simulations and conclusions representative of the state of the art in this type of application, it is necessary to re-emphasize some of the more interesting aspects that will subsequently allow further direct application of the material presented in this thesis.

First, it is recalled that the investigation was exclusively written using non-classified materials which doesn't allow the thesis to present the latest and up to date materials on the subject.

Similarly there exist other areas, such as Inertial Navigation that would logically be part of the overall study but due to their scientific strategic and tactical values, security classifications could not be pursued.

Nevertheless, the data and the material compiled serve as a basis for more detailed studies that will allow further development of the existing systems.

It is my personel belief that the areas which require, for the present, the greatest emphasis is the area of automatic propulsion because it is precisely this area that can immediately lead to initial reduction in manned personnel in a war ship. Of all the propulsion systems mentioned, it is the steam system which would require the

greatest study and greatest effort to improve the actual system. For instance, one of the largest areas requiring further study would be the automatic light-off and securing of a steam system which hasn't yet been realized because with the actual techniques, the added expense does not seem to warrant the added benefit. One initial solution to this problem could take the shape of utilizing the already existing automatic controls, to assist the operator in the manual operation.

From the elementary point of view of reducing the ship to a platform to utilize the weapons the greatest effort of automatic control should be in the improvement of the roll stabilization. In this aspect, much has been done with the use of fins but the greatest theoretical advantages that would be obtained using gyroscopic systems require a further and intensive effort.

Lastly, I believe that in the actual situation, there is great preoccupation at all levels of opinions, concerning the problems of contamination in their different aspects, shall in the near future force the government and industry to spend enormous sums of money for investigation to combat those problems. Without a doubt, the related naval application will be the control of contaminated ballast water, where a great deal of effort should be applied because in reality, the present proposed systems are not a practical and effective solution.

## BIBLIOGRAPHY

### Chapter II


Evans, Rowland Grayson, Comparative Analysis of Steering Proposals, June 1966.

Butterfield, A. M., "Steering Gears Past, Present, and Future Types," Paper, New England Section, SNAME, June 1953.

"New High Pressure Ship's Steering Gear," Shipping World and Shipbuilder, 1 July 1965, pp. 84-88.

Taplin, A., "Notes on Rudder Design Practice," DTMB Report 1461, October 1960, pp. 127-149.

Eda, H. and Crane, C. L., Jr., "Steering Characteristics of Ships in Calm Water and Waves," Paper No. 3, SNAME, November 1965.

"Military Specification, Steering Gears, Electro Hydraulic Marine," MIL-5-1780BD, Art. 3.2.1, 21 April 1960.

Den Hartog, J. P., "Advanced Strength of Materials," McGraw-Hill, New York, 1952.

Eshbach, D. W., "Handbook of Engineering Fundamentals," Second Edition, J. Wiley and Sons, New York, 1952.

"Vickers Hydraulics," Bulletin 5001-E, Vickers Incorporated, Troy, Michigan, May 1965.

Hilderbrand, F. H., "Methods of Applied Mathematics," Prentice Hall, New York, 1952.

Roark, R. J., "Formulas for Stress and Strain," McGraw-Hill, New York, 1943.

Hinsch, K., "Recent Developments in Hydraulic Rotary Vane Steering Gear," AEG Progress, Berlin, Germany, nl, 1957.

Nickerson, A. M., Jr. and Olson, V. A., "Steering Gears and Their Selection," Trans. SNAME, Vol. 60, 1952, p. 504.

Dudley, D. W., "Gear Handbook," Mc Graw-Hill, New York, 1962.

Raven, F. H., "Automatic Control Engineering," McGraw-Hill, New York, 1961.

Gibson, J. E., "Nonlinear Automatic Control," McGraw-Hill, New York, 1963.

Jacobs, Winnifred R., "Methods of Predicting Course Stability and Turning Qualities of Ships," March 1963, U072595 Davison Lab. Report 945.

Eda, H., "Low Speed Contrability of Ships in Wind," October 1966, U105506 Davison Lab. Report 1162.

Brunk, D. H., "A Semiautomatic Steering Device for the M.S.O.," September 1965.

Chen, Hsao-Hsin, "The Experimental Determination of the Rudder Forces and Rudder Torque of a Mariner Class Ship Model," July 1967, U110831.

Dewhrst, Peter K., "Experience in the Control of Ships by Right Angle Drive Trusters," Proceedings 2nd. S.C.S.S.

Sleiertin, Robert A., "A Concept for Automatic Steering of Large Oceangoing Barges," Proceedings 2nd. S.C.S.S.

Norman, David W., "Jet Flaps and Jet Assisted Rudders for Ship Control," Proceedings 2nd. S.C.S.S.

Folsom, David Lindley, "Stability of Autopiloted Ships in a Following Sea," May 1967.

Crane, C. Lincoln, "Studies of Ship Maneuvering Response to Propeller and Rudder Actions."

## Chapter III

Research and Development Report AD-854 790L 13/10 13/2 Naval Ship Research and Development, Lab Analysis Shipboard Oil-Pollution Control System for Ballast and Bilge Waters.

Harbor Pollution by Liquid Wastes Discharges and Summary of Recent Efforts to Eliminate Pollution.

Comparative Evaluation of Four Methods for Detecting Minute Quantities of Free Water in Hydrocarbon Fuels.

## Chapter IV

Kiss, Ronald A., Full Scale Test of Passive Roll Stabilization Tank Installed Aboard Ocean Survey Ship OSS 02, Proceedings Second Ship Control Systems Symposium.

Bowles, R. E., and Meherenz, A. F., Fluidic Roll Stabilization System, Proceedings Second Ship Control Systems Symposium.

Webster, William, Analysis of the Control of Activated Antiroll Tanks.

The Society of Naval Architects and Marine Engineers Transactions, Vol. 75 (SNAME).

Newman, J. N., "Damping and Wave Resistance of a Pitching and Heaving Ship," Journal of Ship Research, Vol. 3, June 1959.

Newman, J. N., "The Linearized Theory for the Motions of a Thin Ship in Regular Waves," Journal of Ship Research, Vol. 5, June 1961.

Stoker, J. J., Water Waves, Interscience Publishers, Inc., New York, 1957.

Weinblum, G. and St. Denis, M., "On the Motions of Ships at Sea," Trans. SNAME, Vol. 58, 1950, pp. 184-248.

Suyehiro, K., "Yawing of Ships Caused by Oscillation Amongst Waves," INA, Vol. 62, 1920.

Korvin-Kroukovsky, B. V., Theory of Seakeeping, SNAME Publication, New York, 1961.

Chadwick, J. H., "On the Stabilization of Roll," Trans. SNAME, Vol. 63, 1955, pp. 237-280.

Chadwick, J. H., "The Anti-Roll Stabilization of Ships by Means of Activated Tanks," (4 parts), Division of Engineering Mechanics, Stanford University, Report 15, December 1950 - June 1951.

Vasta, J., Giddings, A. J., Taplin, A., and Stilwell, J. J., "Roll Stabilization by Means of Passive Tanks," Trans. SNAME, Vol. 69, 1961, pp. 411-460.

Stepanoff, A. J., Centrifugal and Axial Flow Pumps, second edition, John Wiley and Sons, Inc., New York, 1957.

Auslaender, J. and Leopold, R., "The Mathematical Generation of a Realistic Sea," Hydronautics, Incorporated Technical Report 001-13, September 1963.

Gall, D. A., "Minimizing Ship Motions," MIT Department of Naval Architecture and Marine Engineering, Report No. 11, 1964.

Truxal, J. G., Automatic Feedback Control Synthesis, McGraw-Hill Book Company, Inc., New York, 1955.

Chadwick, J. H., On the Stabilization of Roll, SNAME, November 1955.

Conolly, J. E., Rolling and Its Stabilization by Active Fins, SNAME, March 1968.

Allen, J. F., "The Stabilization of Ships by Activated Fins," INA, Vol. 87, 1945, pp. 123-149.

Bassett, P. R. and Hodgkinson, F. P., "New Studies of Ship Motion," Trans. SNAME, Vol. 43, 1935, pp. 286-295.

Gover, S. C., "Observations on Ship Motions at Sea," Proceedings of First Conference on Ships and Waves, Council on Wave Research and SNAME, 1954.

Price, David, Gyroscopic Stabilization of Ship's Roll, Proceedings Second Ship Control Systems Symposium.

Comstock, Principles of Naval Architecture, SNAME 1967, pp. 687-689.

Chapter V

Rubis, C. J. and Moken, A. J., "Naval Ship Research and Development Center," Report 2846, Gas Turbine Ship Propulsion Dynamics, January 1969.

Engineering Laboratory, Annapolis, Report 21402.

Banham, J. W., "Analysis of Automatic Control System Design for Main Propulsion Machinery," Naval Boiler and Turbine Laboratory, Project B-452-I.

Spencer, J. B., "Analysis and Simulation of Main Machinery Pneumatic Control Systems Showing How Optimum Control Can Be Obtained.

Littman, Morris M., "The U.S. Navy's First Automated Steam Plant.

Zavod, P. H., "Simulation and Development of an Automatic Shaft Speed Control System.

Seelinger, J., and Billock, W., "Merchant Ship Automation, Past, Present and Future.

Boatwright, G. M. and Weinert, E. P., "Combined Power Plants," SNAME, March 20, 1964.

Mudd, Donald G., "Digital Hybrid Control of Conventional Propulsion Systems," U.S. Navy Electronics Laboratory, San Diego, California.

Cocks, R. E., "Propulsion Machinery Control Practice in the Royal Navy," Proceedings S.C.S.S., Vol. 2.

Clayton, Curtis T. and Morris, George W., "Design of Automatic Control and Monitor System for a Multiple Diesel Unmanned Mach-Plant.

Thaler, George J. and Vinroot, C. A., "Cumbustion Control System."

Banhan, J. W., Jr., and Zavod, P. H., "Experimental Determination of Open Loop Frequency Response Characteristics of DLG-6 Class Steam Generator System," Naval Boiler and Turbine Lab., Report of NBTL RDT Project B-481, F013-06-06 Task 4182, March 1963.

Banham, J. W., Jr., "Development of Simplified Combustion Controls for CVA 66," Naval Boiler and Turbine Lab.., Report of NBTL RDT Project B-481,F013-06-06, Task 4200, June 1962.

Brown, D. H., "Machinery Systems and Control Considerations," General Electric Report 269 402, December 1961.

Banham, J. W., Jr., "Evaluation of Pre-prototype Pure Fluid Boiler Combustion and Feedwater Control System," Naval Ship Engineering Center Philadelphia Division, Project B-584 11 S-F0130606 Task 4200, December 1966.

Marron, Hugh W., "U.S. Navy Gas Turbine Powered Ocean Escort," Naval Engineers Journal, August 1969.

Beckjord, E. S., "The Stability of Two-phase Flow Loops and Response to Ships Motion," Sept. 1960.

Martinelly and Nelson, "Prediction of Pressure Drop During Forced Circulation of Boiling Water," ASME Transaction, Vol. 70.

Frey, G. W. and Wilson, K. G., "The Development of Bridge Propulsion Control for Turbine Driven Vessels."

Tawaii, Edward, "Realiability and Maintainability Analysis for Proposed Steam and Gas Turbines," Naval Applied Science Lab Brooklyn, N.Y., Project NASL-920-72, AD815 451L,

Donnelly, J. W., "Development and Evaluation of Boiler Smoke Density Optimizer, Naval Ship Engineering Center, Philadelphia, AD 662 415, Nov. 1967.

Naval Ship System Command, Washington, "Hydraulic Servo Control Valves," AD 864 255L, Dec. 1969.

Amelink, H., "Automation and Remote Control on Ships. A Selective Bibliography," AD 832 259, Feb. 1968.

Neff, T. and Mudd, D. G., "Digital Control of a Naval Steam Generating System," U.S. Navy Electronics Lab., NEL/ Report 1461, May 1967.

Dusinberre, George Merrick and Lester, John Campbell, "Gas Turbine Power."

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13. ABSTRACT

An analysis of the problem of Naval Ship Automation is made. The purpose is to search the nonclassified existing documents and to present a general view of the state of the art in Naval Ship Systems. This study covers material that is concerned with conventional Naval Surface Ships, such as destroyers and frigates. Fire Control Systems are not considered because such studies already exist at the present time.

| KEY WORDS | LINK A | | LINK B | | LINK C | |
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| DISCHARGE CONTAMINATION | | | | | | |
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